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REVISION 4

NAVAL SHIPS' TECHNICAL MANUAL

CHAPTER 231

PROPULSION AND SSTG STEAM TURBINES

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CHAPTER 231

PROPULSION AND SSTG STEAM TURBINES

SECTION 1
INTRODUCTION

231-1.1 STATEMENT OF PURPOSE

231-1.1.1 GENERAL. This chapter discusses the design, operation, and maintenance of naval steam turbines. These include propulsion (main) turbines, ship service turbine generator (SSTG) turbines, and coolant turbine generator (CTG) turbines. This chapter may be used for training or general information. Unless otherwise specified by other contractual documents, the procedures in this chapter are not mandatory.

231-1.1.2 PROPULSION TURBINE DETAILED INFORMATION. Propulsion turbine details are contained in the manufacturers' technical manuals. The technical manuals provide most of the information routinely needed, although referring to detail drawings may occasionally be required. MIL-T-17600, Turbines, Steam, Propulsion Naval Shipboard, is the general military specification for manufacturing propulsion steam turbines.

231-1.1.3 SSTG DETAILED INFORMATION. Refer to the manufacturers' technical manuals and detailed drawings when inspecting or repairing SSTG turbines. MIL-T-24398, Turbine, Steam and Reduction Gear, Auxiliary, Generator-Drive (Naval Shipboard Use), is the general military specification for manufacturing SSTG turbines. MIL-T-17523, Turbine, Steam, Auxiliary (and Reduction Gear) Mechanical Drive, was the previous military specification used on many earlier SSTG contracts.

231-1.1.4 GENERAL TURBINE WORK. For any turbine work, use the documents referred to in the overhaul work package. In the absence of specific documents, use the manufacturer's technical manual or detailed drawings.

231-1.1.5 TRAINING. Attending Navy training courses and studying other Navy publications will aid in understanding the thermodynamics of turbine operation and how the turbine fits into the propulsion system as a whole.

231-1.1.6 DEVIATIONS. Deviations from approved, authorized procedures should be requested from the Type Commander (for example, SUBLANT, SURFPAC); Type Desk (for example, Naval Sea Systems Command (NAVSEA) PMS 393, 312, etc.); or Life Cycle Engineer, NAVSEA 03Z23. See paragraph 231-6.1.3.3.

231-1.2 RECOMMENDED CHANGES

231-1.2.1 This chapter has been extensively revised to update information and cover the important aspects of steam turbine management. Liberal constructive criticism in any aspect, such as detail of coverage, subjects, and level of difficulty, is invited. Comments and recommended changes should be referred to NAVSEA in accordance with NSTM Chapter 001, General - NSTM Publications Index and User Guide .

231-1.2.2 The instructions, procedures, and recommendations of this chapter complement, and are in general agreement with, those of the turbine manufacturer. If instructions in this chapter and those of the manufacturer's technical manuals conflict, NAVSEA should be notified by a Technical Manual Deficiency Evaluation Report (TMDER) to reconcile differences.

231-1.3 SPECIFICATION REVISIONS

231-1.3.1 The specifications and instructions referred to in this chapter have not been given a revision letter. Use the contractual or latest revision applicable.

SECTION 2

STEAM TURBINE DESIGN

231-2.1 PROPULSION TURBINES

231-2.1.1 TYPES OF PROPULSION TURBINES IN NAVAL SERVICE. Several turbine types are used for propulsion service in naval ships. The two principal steam turbine ship propulsion plants in use today are the geared-turbine drive and the turboelectric drive. The direct-drive turbine propulsion plant, which was the forerunner of the present-day plants, was found on only USS JACK (SSN 605) and USS NARWHAL (SSN 671). The submarine turboelectric drive propulsion plant was found only on USS LIPSCOMB (SSN 685).

231-2.1.1.1 General. Steam turbines must operate in a relatively high range of revolutions per minute (RPM) to obtain the maximum amount of work per pound of steam used. Propellers, however, operate most efficiently in a much lower RPM range. The need for suitable means of operating both the turbine and the propeller within their respective efficient RPM ranges led to the development of the geared-turbine drive and the turboelectric drive.

231-2.1.1.2 Reduction Gear Unit. In the geared-turbine drive a reduction gear unit is the means for reducing high turbine RPM to the much lower propeller shaft RPM. In the direct-connected electric drive the necessary speed reduction is brought about electrically. That is, the turbine drives a generator at high RPM, and the generator furnishes power for an electric propulsion motor that operates at low RPM. The turbine installations in these two drives differ considerably, as will be pointed out in the succeeding paragraphs.

231-2.1.1.3 Geared-Turbine Drive. In the geared-turbine drive the parts or sections that make up the individual propulsion units consist of the main turbines and the reduction gear (Figure 231-2-1 and Figure 231-2-2). No active U.S. ships have geared-turbine plants with a cruising turbine. Cruising turbines are obsolete and were used in only smaller ships, frigates and destroyers, of World War II vintage, such as the DD 692 class.

231-2.1.1.3.1 The propulsion unit has two astern elements, one at each end of the low-pressure (LP) turbine, which is typical for combatant ships. Auxiliary ships usually have one astern element instead of two, and that one is invariably located at the forward end of the LP turbine (the end farthest from the reduction gear).

231-2.1.1.4 Turboelectric Drive. Unlike geared-turbine propulsion plants, which have one or more ahead turbine elements and one or two astern elements for each propulsion shaft, the turboelectric drive installations have a single turbine unit for each installed shaft on surface ships (Figure 231-2-3). The surface ship turboelectric propulsion unit includes a turbine, main generator, propulsion motor, direct current generator for supplying excitation current to the generator and the propulsion motor, and a propulsion control board. The only surface ships that have turboelectric drive are USNS VANGUARD and USNS REDSTONE, which are part of the Military Sealift Command.

231-2.1.1.4.1 The submarine turboelectric propulsion unit differs from the surface ship unit in that the design incorporates two turbines and two generators driving a single propulsion motor.

231-2.1.2 TURBINE DEFINITIONS

231-2.1.2.1 Single-Flow Turbine. In a single-flow turbine, the steam is confined to a single blade path as it expands through the turbine (Figure 231-2-2).

231-2.1.2.2 Double-Flow Turbine. In a double-flow turbine, the steam divides and flows through two blade paths (Figure 231-2-2).

231-2.1.2.3 Impulse Element. In a typical condensing naval steam turbine, one or two impulse elements are located in the high-pressure (HP) area closest to the nozzle block (Figure 231-2-4). The impulse element has no pressure drop across the rotating blades, only across the nozzle(s).

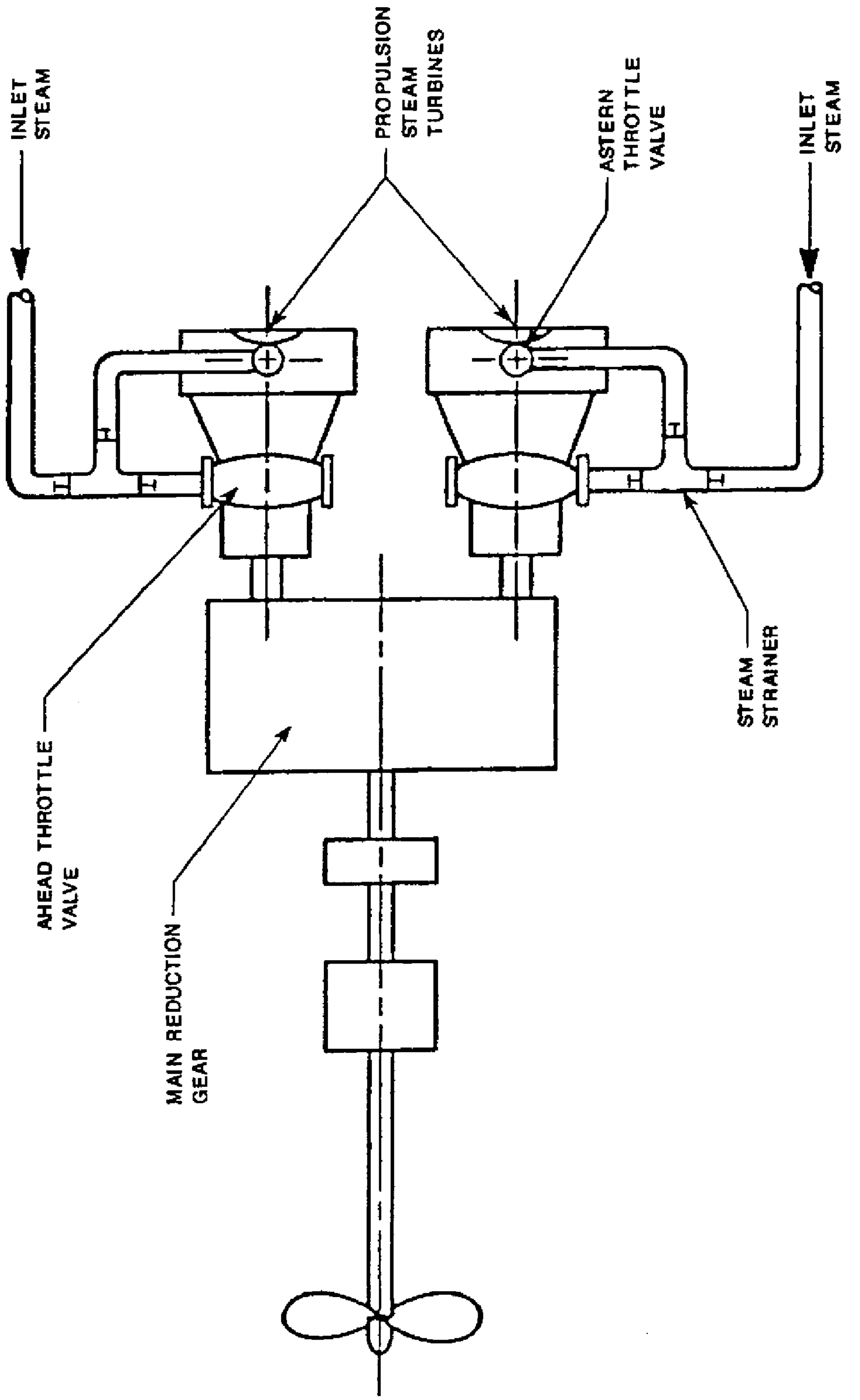


Figure 231-2-1 Typical Main Propulsion Steam Turbine Arrangement for Submarines

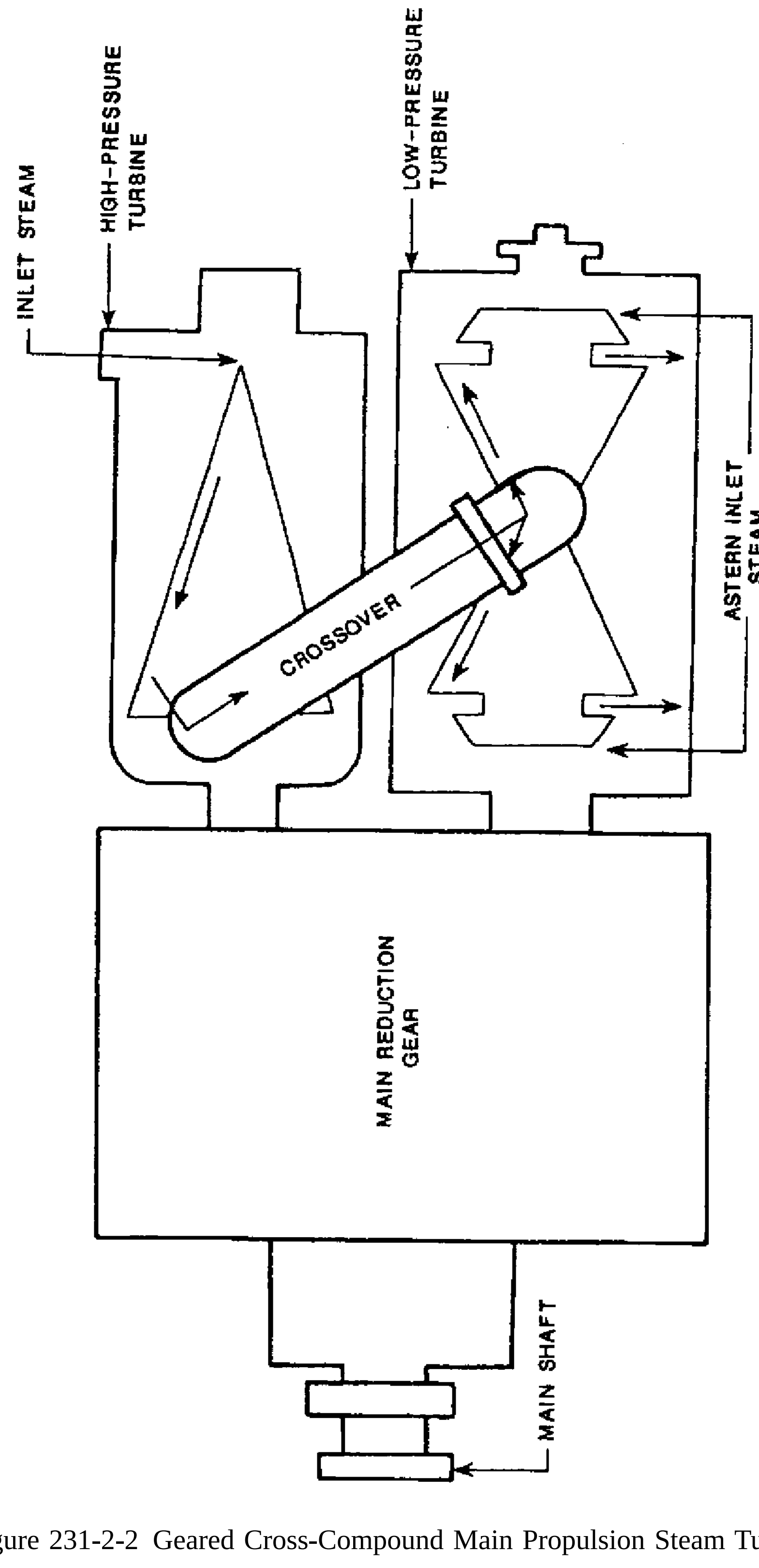


Figure 231-2-2 Geared Cross-Compound Main Propulsion Steam Turbine

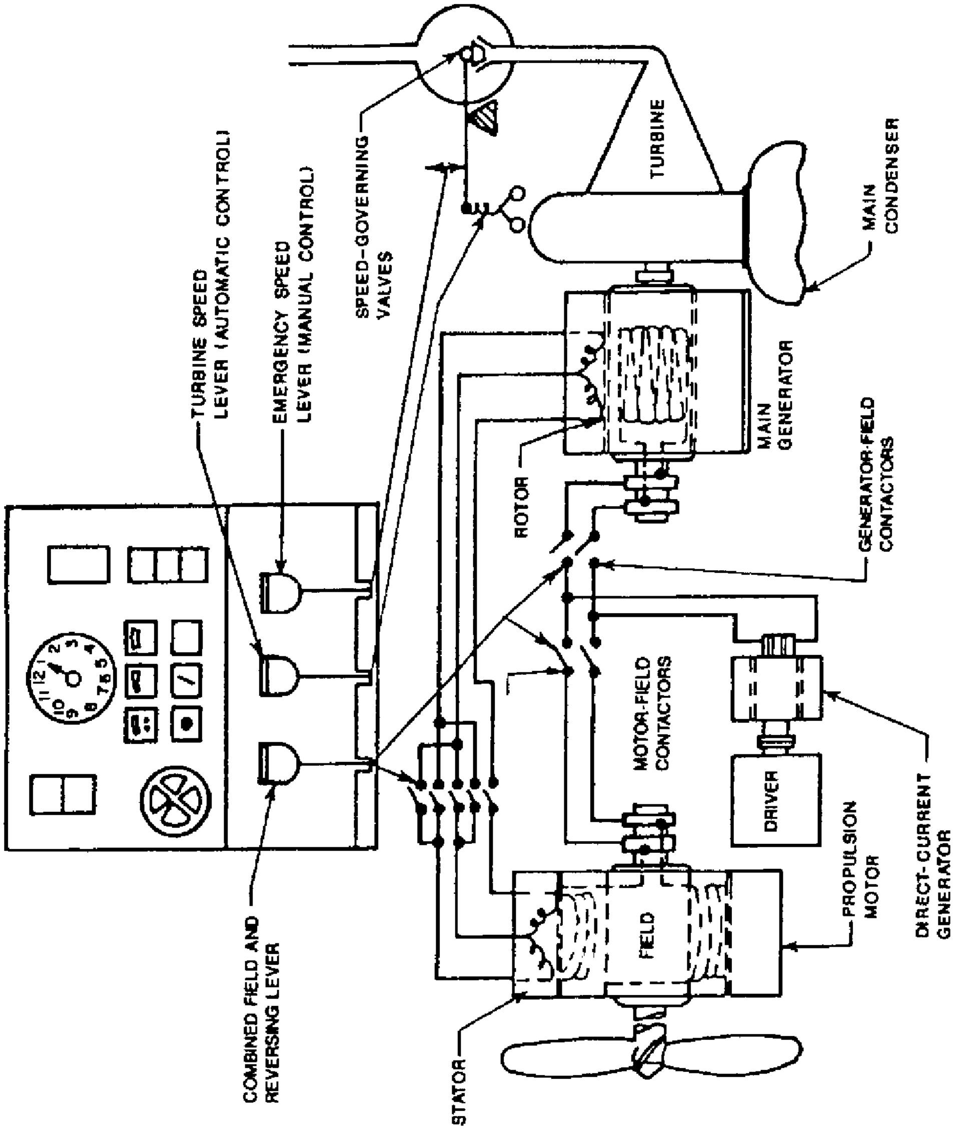


Figure 231-2-3 Diagram of the Turboelectric Drive for Surface Ships

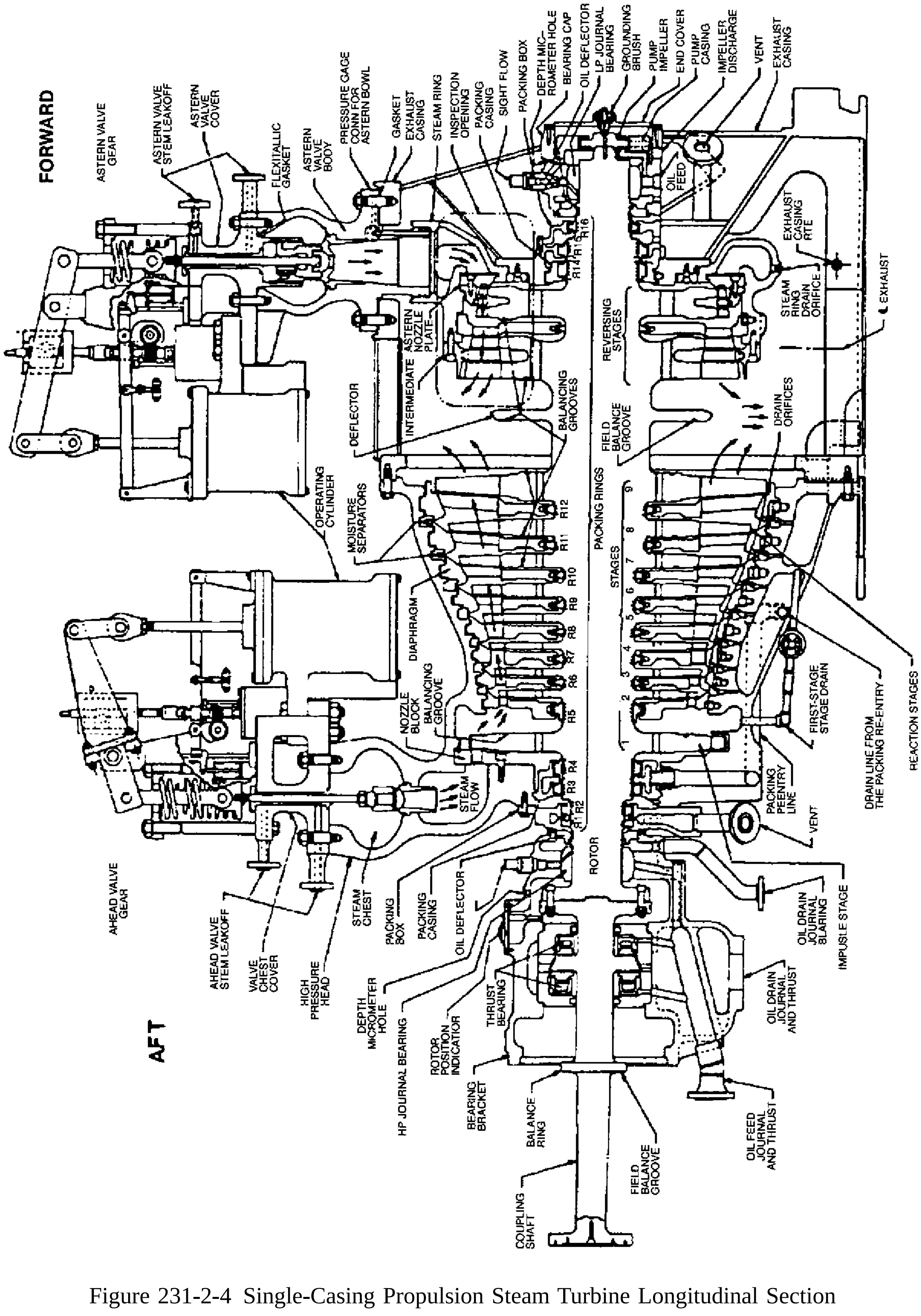


Figure 231-2-4 Single-Casing Propulsion Steam Turbine Longitudinal Section

231-2.1.2.4 Reaction Element. In a typical condensing naval steam turbine, many rows or stages of reaction blades are located toward the LP area closest to the exhaust area (Figure 231-2-4). In a reaction element, pressure drops in each stage are divided equally between stationary and rotating blades.

231-2.1.3 TYPES OF UNITS

231-2.1.3.1 Single-Casing Unit. The single-casing unit consists of one HP ahead element and an astern element in one casing (Figure 231-2-4). Usually, two of these units drive one reduction gear, as on submarines.

231-2.1.3.2 Straight-Through Unit (Cross-Compound Set). The straight-through unit consists of two ahead elements, known as an HP element and LP element. The HP and LP elements are contained in a separate casing and are commonly known as the HP and LP turbines, respectively. The HP and LP turbines deliver power to a single shaft through a gear train and are coupled separately to the reduction gear. Steam is admitted to the HP turbine, flows straight through the turbine axially without bypassing any stages (there is partial bypassing of the first row of blades at high power), and then is exhausted to the LP turbine through a crossover pipe. The LP turbine can be either single or double flow.

231-2.1.3.3 External Bypass. The external bypass propulsion unit (CGN applies) is similar to the straight-through unit except that at high powers main steam is admitted through separate chest valves to stages downstream of the HP first stage.

231-2.1.3.4 Internal Bypass. The internal bypass propulsion unit (DLG applies) is similar to the straight-through unit except that at powers above the specified cruising power steam bypasses (valved) the HP first-stage shell to the downstream stages.

231-2.1.3.5 Series-Parallel Unit. The series-parallel unit consists of three ahead elements, known as the HP, the intermediate pressure (IP), and the LP elements. The HP and IP elements are combined in a single casing and are known as the HP-IP turbine. Steam is admitted to the HP-IP turbine and exhausted to the LP turbine through a crossover pipe. For powers up to the most economical point of operation, only the HP element receives inlet steam, with the IP element being supplied in series with steam from the HP element exhaust. At powers above this point of operation, both elements receive inlet steam in a manner similar to that in a double-flow turbine. During ahead operation no ahead blading is bypassed. Series-parallel units are used on such combatant vessels as CVA's, FF's, CG's and DDG's.

231-2.1.3.6 All these propelling units contain an astern element for backing or reversing. An astern element is located in each end of a double-flow LP turbine casing or can be in either end of each single-casing turbine or single-flow LP turbine.

231-2.2 SHIP SERVICE TURBINE GENERATOR (SSTG) TURBINES

231-2.2.1 Most SSTG turbines in naval service are straight-through condensing units that either drive through a gear or are directly connected to a generator. CVN 65 has a noncondensing SSTG. Surface ship SSTG's use gearing, while submarine SSTG's are direct drive. The turbine is designed for axial steam flow with multiple stages that use a combination of impulse and reaction staging. The SSTG is classified by its power output capacity in kilowatts. A typical SSTG unit in use today is shown in Figure 231-2-5.

231-2.3 TURBINE CONTROLS

231-2.3.1 CONTROLS. In this context the word ″controls″ refers to devices used to adjust the quantity and direction of steam flowing through the turbine sections.

231-2.3.2 THROTTLE OR MANEUVERING VALVES. The single-throttle or maneuvering valve method of controlling turbine steam admission is essentially outmoded but is still in use in propulsion turbines in some old ships. With throttle or maneuvering valves, all steam entering the turbine flows through the maneuvering (throttle) valve and speed is varied by opening or closing this valve. Large throttling losses are avoided through the greater portion of the power range by varying the number of steam chest nozzles opened downstream by separately controlled hand-nozzle valves.

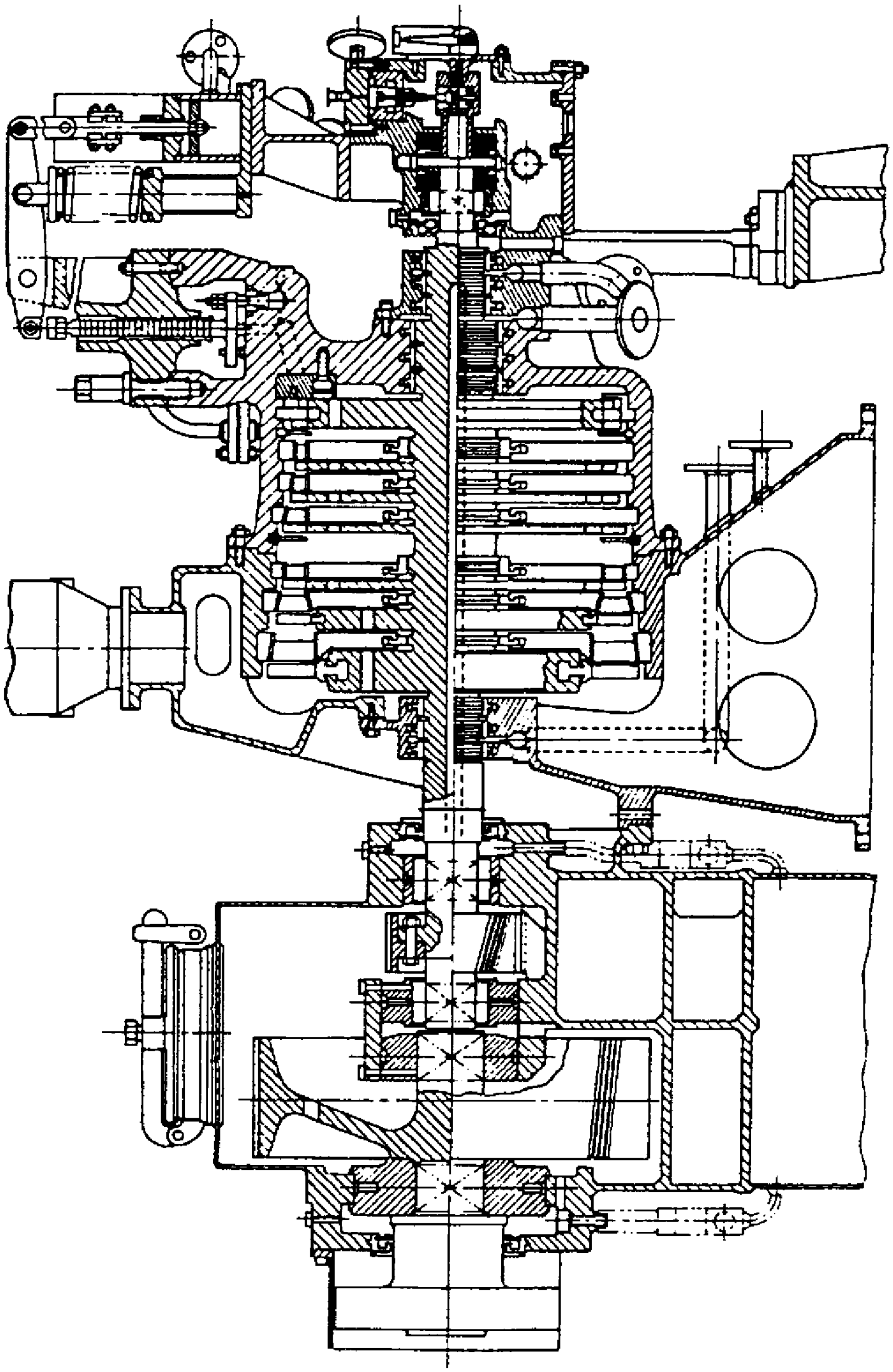


Figure 231-2-5 Longitudinal Section of 1500-kW SSTG Turbine and Gear

231-2.3.3 CONTROL VALVES. A control valve transfers steam from the turbine steam chest to the inlet area of the first-stage nozzles in the quantity required to produce the desired power level. These valves are controlled by several methods, as described in paragraphs 231-2.3.3.1 and 231-2.3.3.2.

231-2.3.3.1 Individual Hand-Controlled Valves. Individual hand-controlled valves are used in conjunction with an upstream maneuvering or throttle valve, as indicated in paragraph 231-2.3.2. They are used only in the full-open or closed positions. Medium speeds are attained by adjusting the maneuvering valve. Since throttling at the maneuvering valve reduces the energy available to the turbine blading, select a combination of hand-controlled nozzles so that the throttle valve is as close as possible to being wide open, during normal steaming. Exceptions to this would be a tactical situation, where more hand valves would be kept open to satisfy demands for rapid and large increases in power, or docking.

231-2.3.3.2 Cam- and Bar-Lift-Operated Control Valves. The maneuvering and individual hand-operated control valve system (developed primarily for merchant service) is still used in older naval ships performing similar duty. Strict maneuverability requirements for combatant and newer fleet support ships, however, have resulted in the universal adoption of a system for full control of all turbine power levels by a single ahead handwheel (ahead throttle) and a single astern hand wheel (astern throttle). Two arrangements are used for controlling the control valves in all newer designs: cam- and bar-lift, with the latter being the more common.

a. Cam-Lift. Individual cam-lift valves are currently used only in larger-size General Electric Company units. In these, a geared-down, single camshaft carrying a cam for each valve is turned by a reach rod connected to a remotely located throttle handwheel. A cam, crosshead, lever, valve stem, and valve arrangement is used for each control valve; sequential opening of control valves is obtained by proper angular staggering of lift cams on the camshaft. A spring in each control valve assembly maintains lever roller contact with the lift cam on opening and provides the valve closing force when the camshaft is moved in the closing direction. Knock-down pins on each lever force the valve in the closing direction if the spring force is inadequate because of derangement or hangup.
b. Bar-Lift. The bar-lift valve arrangement is less complex. It consists of a horizontal bar with a series of vertically bored holes, one for each control-stem valve. The bar itself is supported and moved in a vertical plane by two lift rods that penetrate the steam chest. Each of the valves, sometimes referred to as poppets, hangs on a button that rests on top of the bar when the valve is open and off its valve seat. In the closed position, the valves are supported by their valve seats and securely held down by steam pressure. Valves are opened sequentially by varying the stem valve length: the shortest stem valve will open first as the bar lifts and the button-to-bar clearance closes. The button contact surface will be either flat or spherical, the latter being used to avoid spinning and cocking. Various mechanisms (screw thread, rotating levers, cam action) are used in the different designs to transform rotary throttle handwheel motion into the rectilinear motion of the bar-lift rods. The bar lowers to close the valves by spring return, by closing the cam, or by simply reversing the motion in the screw thread drives.

231-2.3.4 TRANSFER VALVES.

231-2.3.4.1 Transfer Valves Performing with Series-Parallel Turbines. In the series-parallel turbine, steam exhausted from the HP element either flows 100 percent in series through the IP element at low powers or is diverted at high powers so that the major portion flows directly to the LP turbine inlet. The minor portion flows in parallel through the IP element to provide for losses, cooling, and a small amount of work output in the HP element. This mode change is accomplished by a crossover valve in conjunction with a swing-check valve. The swing-check valve (located in the line from the HP exhaust to the lower IP inlet) closes automatically when the crossover valve opens because opening the HP exhaust to the LP inlet imposes a pressure difference on the check valve. This pressure difference is equal to the pressure drop across the IP element.

231-2.3.4.1.1 No venting or cooling of the HP element is necessary because 20 to 25 percent of the inlet steam flows through it at full power. That is, the HP element contributes to the shaft output at all powers.

231-2.3.4.2 Transfer Valves Performing with Internal Bypass Turbines. In the internal bypass turbine, one section of stages is bypassed at ship speeds above 25 knots. By providing a parallel valve path, steam either flows through all stages of the HP turbine or is bypassed through internal passages in the casing. This design differs from the external bypass type in that steam is taken from the first-stage shell area rather than from the steam chest. The bypass valve itself is sized, when fully open, to maintain an adequate pressure drop across the bypassed stages to make those stages self-sustaining. That is, power developed just offsets losses.

231-2.3.4.3 Transfer Valve Performance in Straight-Through and Series-Parallel Turbines. Both the series-parallel and the bypass turbines are current designs that take advantage of the lower steam rates and higher horsepower/pound of turbine allowed by the 1,200-psig, 950°F steam system. The bypass or crossover valves, as applicable, have four common attributes that simplify operation:

a. These valves are stroked automatically, either directly through the control mechanism or indirectly through a pneumatic power cylinder.

b. The valves open quickly to minimize the range over which throttling losses occur.

c. The mode of operation of the valves is automatically changed (series to parallel or straight through to bypass) without abnormal speed fluctuations or disproportionate changes in steam demand.

d. At a ship speed of approximately 27 knots, the valves produce a turbine mode change that avoids alterations in normal cruising speed ranges and coincides with the addition of more boilers on line.

231-2.3.5 EXTRACTION VALVES. Some turbines are fitted with extraction openings in the HP or LP turbine elements. Steam cycle efficiency can usually be improved by using steam extracted from the propulsion turbine to heat feedwater or to provide LP steam for other functions. The extraction valves normally open or close automatically to maintain a fixed pressure in the system being supplied with steam. Nonreturn valves in series with the extraction valves prevent steam from backing up into the turbine from the connected system.

231-2.3.6 CONTROL HANDWHEELS. The ability of a ship to maneuver normally and under emergency conditions depends partly on the control system provided to vary turbine power. Although the capability of immediately opening control valves would be desired for maximum maneuverability, the rate is limited by the response of the connected steam supply system.

231-2.3.6.1 General. The maximum force required to operate the throttle handwheel is specified in the design and represents a compromise that allows needed maneuverability within the capability of the equipment. With the forces and turns so limited, reversals from full ahead to full astern and vice versa can probably be made in 45 seconds or less. Typical values for normal hydraulic operation without excess binding in gearboxes are shown in Table 231-2-1. Perform maintenance requirements for lubricating the gearboxes.

Table 231-2-1 THROTTLE HANDWHEEL TURNS

		Maximum Torque (Foot-Pounds)		Number of Handwheel Turns	
Handwheel	**Size (Inches)**	**Submarine**	**Surface**	**Submarine**	**Surface**
Ahead	24	20	40	10 to 15	10 to 35
Astern	18	15	30	6 to 15	12 to 40

231-2.3.6.1.1 Required Torque. There is a peak handwheel torque associated with lifting a nozzle-control (and other) valve against steam pressure. The torque required falls off as the valve is farther lifted, reaching a minimum when the valve is fully open.

231-2.3.6.1.2 Excess Torque. When the peak force exceeds specification value, the force is reduced by:

a. Inserting a torque booster gearset between the throttle and control valve lifting mechanism, thus reducing torque required throughout the range
b. Adding a pneumatic power operator on the transfer or bypass valves to reduce an unacceptable torque peak
c. Reducing friction in reach-rod systems or control-valve linkages. The planned maintenance system (PMS) directs the lubrication of control fittings. Plugs may be replaced with grease fittings as necessary

231-2.3.6.1.3 When troubleshooting an excess handwheel torque problem, first remove steam from the throttles. Mark the throttle control system and disconnect from the throttles. Evaluate the torque required for each throttle handwheel separately, comparing it with other turbine throttles and Table 231-2-1. Continue to isolate throttle system components. Often the problem is found in a throttle control linkage or gearbox misalined or unlubricated.

231-2.3.6.2 Jammed Throttle. When closing an ahead or astern valve, do not jam the handwheel mechanism into the stop bumper. Closing until the resistance of the stop is felt is sufficient. If throttle control mechanisms are jammed into a stop, they can later resist opening and spring open uncontrollably, putting way on the ship.

231-2.3.6.3 Closed Overtravel. Throttle handwheel mechanisms turn one to two handwheel wheel turns in the close direction after the steam valves are shut. This is called closed overtravel and ensures that the steam valves are shut. Too much overtravel would slow throttle response to an emergency. Some throttle handwheels have a remote operating system or turbine throttle hydraulic system. If they are suspected of being jammed, use the handwheels closest to the turbine. This will help prevent unwanted overtravel of the throttles caused by a release of energy stored in the remote operating system.

231-2.3.6.4 Manual Override. Override is provided in ships with power assisted on the throttles or where loss of control oil would cause the throttle valve to trip. The override is accomplished by simply continuing to move the throttle (either ahead or astern) in the open direction. This override feature is an emergency device and should not be routinely used to negate protective devices of the speed-limiting and overspeed trip systems. Since manual override mode is used without oil power assistance, handwheel torques will be higher than those shown in Table 231-2-1. Troubleshooting excess throttle handwheel torques should include disconnecting the turbine from remote handwheels and observing torque at handwheels on the turbine. Also, gearboxes and connections in the remote handwheel system should be checked for proper alinement, adjustment, and lubrication. Remote throttle handwheels with concentric shafts can jam each other, causing excess torque to the other handwheel. Therefore, never jam closed a handwheel of this design, because the internal shafts and gearing may move out of position, causing friction to the other handwheel.

231-2.3.7 TURBINE SPEED-LIMITING CONTROLS. Propulsion turbines that can be accidentally unloaded are always fitted with speed limiters. Accidental unloading might occur by unclutching from the propeller shaft in geared drives, when the turbine is remotely located (bridge control of automated ships), or if arrangements are such that rapid action cannot be taken in case of shaft failure or propeller out of water. Almost all speed-limiting systems use hydraulic oil as an actuating fluid. This actuating fluid transmits an oil pressure that varies with speed, caused by a pump impeller, to a hydraulic cylinder. This hydraulic cylinder moves control valves to positions that will limit speed to approximately 110 percent of design speed. Because the hydraulic systems are complex, the applicable technical manuals detail their operation and checkout. These manuals should be studied in detail.

231-2.3.8 SSTG TURBINE GOVERNOR. All SSTG turbines in naval service are equipped with a constant-speed (speed-regulating) governor. A constant-speed governor is a governor that, by controlling and regulating the steam admitted to a turbine, automatically maintains the speed of the turbine at a predetermined rate, under all conditions of load and exhaust pressure, within the turbine's design limits. Governors of this class are either the centrifugal (direct-acting) or hydraulic-relay type centrifugally controlled or electric speed (electronic) and load sensing, using lubricating oil as the relaying medium for the actuating force. These governors can be operated as speed-droop governors (share load between SSTG's by changing speed). For more information on the governor design refer to the manufacturer's equipment manual. Turbines for driving small turbine generators are usually equipped with constant-speed governors of the centrifugal, direct-acting type. Large generator turbines are usually equipped with hydraulic-relay, constant-speed governors. Refer to paragraph 231-8.20.6 for governor troubleshooting procedures.

231-2.3.9 OVERSPEED LIMITER. Propulsion turbines that can be inadvertently unloaded by such things as uncoupling of the high-speed pinion coupling or loss of propeller, causing an overspeed condition, may be fitted with an overspeed limiting system (overspeed limiter) that uses lubricating oil as the operating fluid to limit the turbine speed. In a single-casing design, which includes the ahead and astern sections, the ahead and astern limiting speeds are normally set at 110 percent of design speed. Refer to the turbine technical manual for actual settings. Note that the overspeed limiting system is inoperable while turbine throttles are in manual override.

231-2.3.9.1 The overspeed limiting system consists of one governor pump and, normally, two overspeed relays: one for the ahead valve gear and one for the astern valve gear. Each relay has an overspeed test valve that allows the operator to simulate an overspeed condition and thereby test the operation of the system.

231-2.3.9.2 The governor pump receives oil from the bearing header. During normal operating speeds the discharge pressure from the governor pump cannot overcome the spring force on the relay piston. During an overspeed condition, however, the output pressure from the governor pump increases, overcomes the spring force, and causes the relay piston to rise. This action results in the lever system lifting the main pilot valve to dump oil from the operating cylinder and start the steam control valves in the closing direction. The valves will not close completely but will hold the speed at the point to which the speed limiting system was set.

231-2.3.9.3 The overspeed limiting system is self-restoring. That is, when the overspeed condition has been corrected and normal turbine speed is resumed, the output pressure of the governor pump will drop. The relay and lever system will then restore the steam control valves to their previous open position.

231-2.3.9.4 The overspeed test valve can simulate an overspeed condition to test the system at any speed above 80 percent of the set speed. By pushing down on or turning the test lever, the overspeed test valve will be raised to allow additional oil pressure (normally blocked off) under the relay piston. Exposing a larger surface area to the governor pump oil pressure causes a larger force to counteract the overspeed limiter spring. The additional pressure on this area is enough to overcome the spring force on the relay piston and lift the piston similar to an overspeed condition. Release of the test lever will restore the system to normal.

231-2.3.9.5 Refer to the manufacturer's technical manual for the detail procedure to follow when setting and testing the overspeed limiter.

231-2.3.10 TURBINE OVERSPEED TRIPS. All turbines that drive through clutches or drive a generator are designed with second-level overspeed protection. The overspeed system normally uses a mechanical speed-sensing element in the rotor that automatically moves on overspeed, opening a valve to dump hydraulic pressure needed to keep the throttle valve open. The trip system can also be triggered manually at any ship or SSTG speed by means of an actuating device at the turbine. Tripping speed is normally between 110 and 112.5 percent of full-power RPM. Test turbine manual trips before and during every startup. Test the turbine overspeed trip as directed by the PMS (typically annually). Do not exceed the maximum RPM given in the turbine technical manual, typically 112 or 112.5 percent of full power RPM. If the turbine has not tripped at the maximum specified RPM, do not continue to operate it until trip is properly set unless ship's force has taken safety precautions (e.g., dedicated personnel at the manual trip, etc.). Tripping of the overspeed trip below the minimum trip speed given in the turbine technical manual is acceptable as long as normal turbine operation is not affected. Do not reset overspeed trip speed without technical assistance from Fleet Technical Assistance (see paragraph 231-6.1.3).

231-2.3.11 TACHOMETER. When the speed-limiting, speed-regulating, overspeed limiter, or overspeed trip settings are checked, use at least two independent means of measuring speed with different principles of measurement. At least one of these measurements should be on a nonstroboscopic tachometer. If component design permits (most propulsion units have a provision for a mechanical tachometer), use a hand-held mechanical tachometer. Do not measure turbine speed with multiple scale portable tachometers.

231-2.3.11.1 Note that speed limiters and overspeed trips are not precision speed control devices. High precision RPM indicators are therefore unnecessary. The first of the two tachometers should be accurate enough to read the RPM within at least 1 propeller RPM. The second tachometer is not used to verify the accuracy of the first, but rather the correct harmonic or speed range. A machine rotating at 4000 RPM will have a harmonic at 2000 RPM. Stroboscopic tachometers have been mistakenly set to this half speed. Turbines would then run at double speed, causing overspeed destruction. An acceptable second tachometer is a vibrating-reed tachometer, which cannot be fooled by harmonics. All speed indicating devices should be correctly calibrated and used by qualified operators. Do not use ship-installed tachometers unless they are periodically and correctly calibrated. When a stroboscopic tachometer is used, compare its reading with an independent speed-measuring device at speeds equal to or less than rated speed before entering the overspeed range.

231-2.3.12 STEAM STRAINERS AND LOW-OIL-PRESSURE ALARMS AND TRIPS. In addition to the protective devices referred to in paragraphs 231-2.3.8 through 231-2.3.10 other safety fixtures are provided on turbines, including steam strainers, and low-oil-pressure alarms and trips.

231-2.3.12.1 A steam strainer is located in the steam line ahead of the control valves to prevent foreign material from entering the turbine. The strainer is a basket made of wire mesh or perforated, corrosion-resistant steel. The steam strainer should be examined and cleaned in accordance with planned maintenance requirements, normally at overhaul or when upstream piping is opened. Investigate the source of any foreign material found. Do not examine original drawing parameters, but verify that the strainer can perform its intended function until the next scheduled inspection.

231-2.3.12.2 The low-oil-pressure alarm contactor is a device provided on most turbines in the lubricating oil system at the most remote bearing that completes an electric circuit to an audible alarm in case the oil pressure drops to the minimum for safe operation.

231-2.3.12.3 The low-oil-pressure trip is a device provided on some turbines and connected to the lubricating oil system. It operates in conjunction with the overspeed tripping mechanism and shuts down the turbine if the oil pressure drops below a predetermined minimum. Low-oil-pressure trips are generally provided on turbine generators equipped with a governing system that will not shut down the turbine in case of loss of oil pressure. Low-oil-pressure alarm contactors are generally not provided on turbines equipped with low-oil-pressure trips. Consult the drawings or manufacturer's instruction manual for the correct setting of a low-oil-pressure alarm contactor or low-oil-pressure trip on any particular turbine. These devices are usually set to operate at about 4 psig. Where both protective devices are installed, the alarm is set 2 to 3 psi above the trip. When a low-oil-pressure trip or alarm contactor operates, stop the turbine and locate and correct the trouble immediately. Subsequent actions that should be taken include:

1. Note and check oil temperature (sight flow indicator).
2. Switch duplex strainer baskets (retain foreign material).
3. Sample lube oil, label, and retain.
4. Take lube oil readings.
5. Check thrust indicator.
6. Verify proper flow lube oil pressure to bearings.
7. Establish cause.

231-2.4 SHAFT AND DIAPHRAGM PACKINGS

231-2.4.1 PURPOSE. Because the turbine rotor penetrates and turns with respect to the casing and because internal pressures differ from atmospheric pressure and vary from stage to stage, sealing is required to prevent leakage. Shaft packings, in conjunction with the gland seal system, are used either to keep steam from leaking out or to keep air from leaking into the turbine. Diaphragm packings minimize internal (stage-to-stage) leakage in impulse stages. Strips that radially seal the shrouding of the reaction blading prevent leakage in reaction stages.

231-2.4.1.1 Steam leaking through the diaphragm packing bypasses a stage or whole turbine section and cannot perform in the active steam path. A throttling loss occurs even if the steam is readmitted at a downstream stage. It is important, in terms of performance, to design for minimum leakage, and it is important to operate equipment in a manner that maintains the small clearances. The precautions to be taken are covered in paragraphs 231-3.2.2 through 231-3.2.2.5, 231-3.5.2 through 231-3.5.2.5.3, and 231-3.10.3 through 231-3.10.3.5.

231-2.4.2 DESCRIPTION AND PRINCIPLE. Shaft and diaphragm packings are either carbon or labyrinth packings. These packing types are discussed in paragraphs 231-2.4.2.1 and 231-2.4.2.2.

231-2.4.2.1 Carbon Packing. Modern propulsion turbines no longer have carbon packing, but it is still in service in propulsion units of many active older ships, particularly battleships. A typical carbon packing gland ring is illustrated in Figure 231-2-6.

231-2.4.2.1.1 A carbon ring normally consists of four or six butted arc segments held together by a circumferential tensioning device. The assembled ring is kept from turning with the shaft by a stop plate or a key between the ring and gland retainer. Rings are set with a cold ring-to-shaft clearance so that when the shaft and ring heat to operating temperature the running diametral clearance is about 0.002 inch. A procedure for setting carbon ring clearances using a mandrel is defined in NSTM Chapter 502, Auxiliary Steam Turbines .

231-2.4.2.1.2 The quantity of steam that the seal will pass is a function of radial clearance. This makes the carbon seal very effective because it can be set closer to the shaft than noncontacting metallic seals and allowed to contact the shaft and run in without damaging the shaft. Leakage is also inversely a function of seal length along the leakage path. Tandem, narrow seals are preferred to single, long seals because steam velocities along the shaft tend to be destroyed by eddying in the spaces (labyrinth) between the rings.

231-2.4.2.1.3 Although the carbon seal is normally considered a radial type, the ability of the seal to seat axially and shut off leaks around and through the seal-to-holder interface is at least equally important and should not be overlooked. The surfaces involved are designed to be square and smooth so that the combination of good fit and steam pressure (which forces the ring downstream) provides effective seating. The carbon rings are limited in service to a temperature of about 500°F and are most commonly used in the relatively cool end glands of LP turbines. Four rings are used in each gland, but they are not effectively in series (paragraphs 231-2.6 through 231-2.6.4).

231-2.4.2.2 Labyrinth Packing. Two forms of labyrinth packing are used in naval turbines. Older packing ring designs have a series of strips made of brass or a similar soft alloy that are machined to a narrow thickness at the tip and individually rolled or caulked into a seal holder. The radial clearance between these seal strips and the shaft varies from 0.010 to 0.025 inch depending on the shaft diameter and operating temperature. Packing rings of later design are illustrated in Figure 231-2-7 and Figure 231-2-8. They are called labyrinth seals because they have multiple seals on one ring. They are designed to seal against a constant uninterrupted diameter with a series of increasing or decreasing steps, or alternately high and low steps that provide maximum eddying in the labyrinth spaces between the seals. These rings are more expensive and initially more difficult to manufacture but offer many advantages that are apparent when considering various design features. These advantages are:

a. The rings are much more rugged and will not lose seal segments or break because of brittleness.

b. A completely machined ring allows more accurate spacing of individual teeth or rings and closer control over radial clearance.

c. Close dimensional control permits high-low step tooth design, which is most effective in limiting leakage in a given packing length.

d. The rings have a spring backing that permits the ring to move, when touched by a temporarily bowed or transiently vibrating shaft, thereby minimizing wear.

e. The design simplifies packing removal, repair, or replacement.

f. Materials can be changed to suit temperature conditions.

g. The hook design (L- or T-shaped) and spring backing radially position the seal in cold conditions, with an additional radially inward seating force provided by the radial difference in steam pressure.

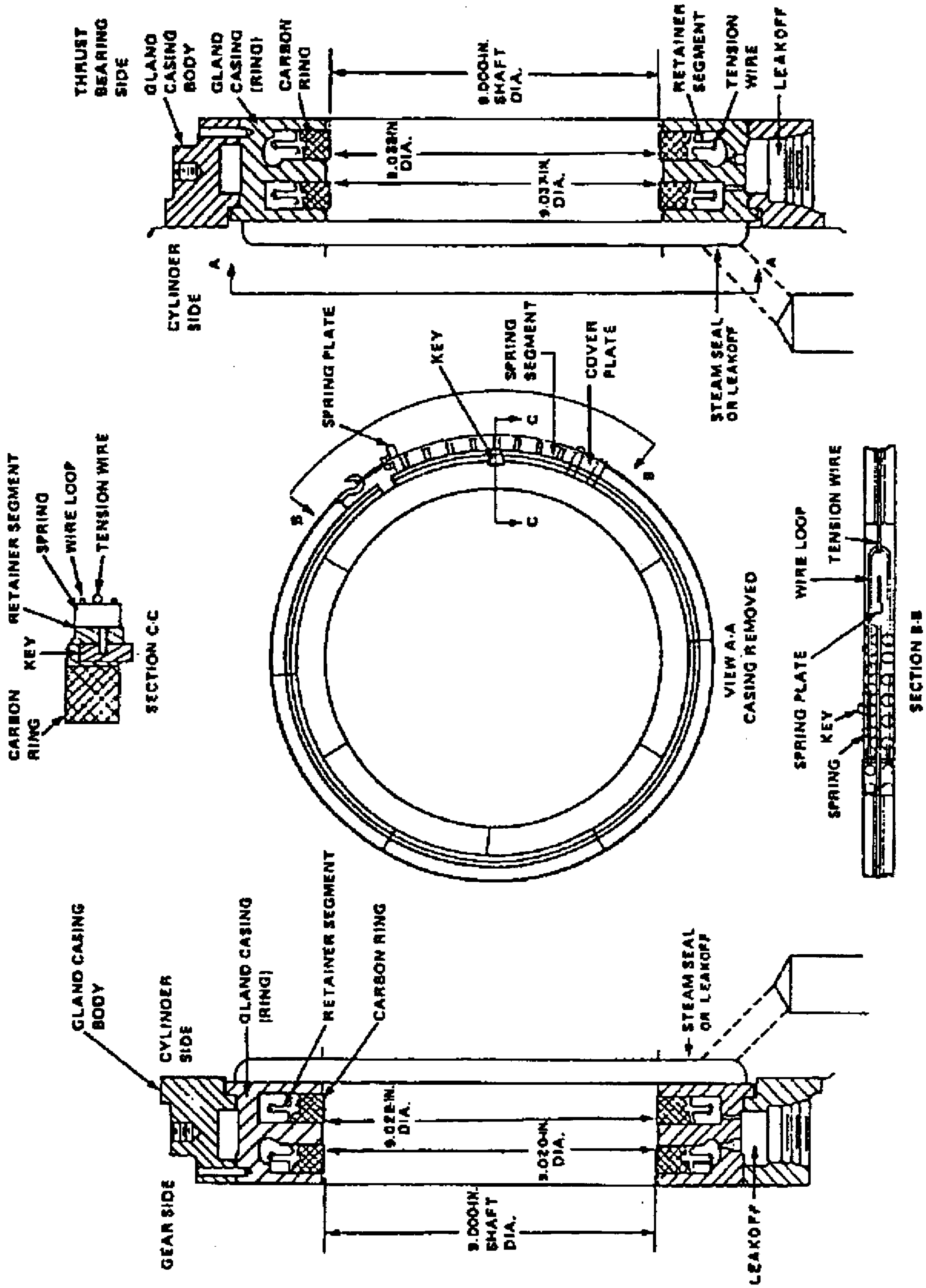


Figure 231-2-6 Shaft Packing Gland Rings (Carbon)

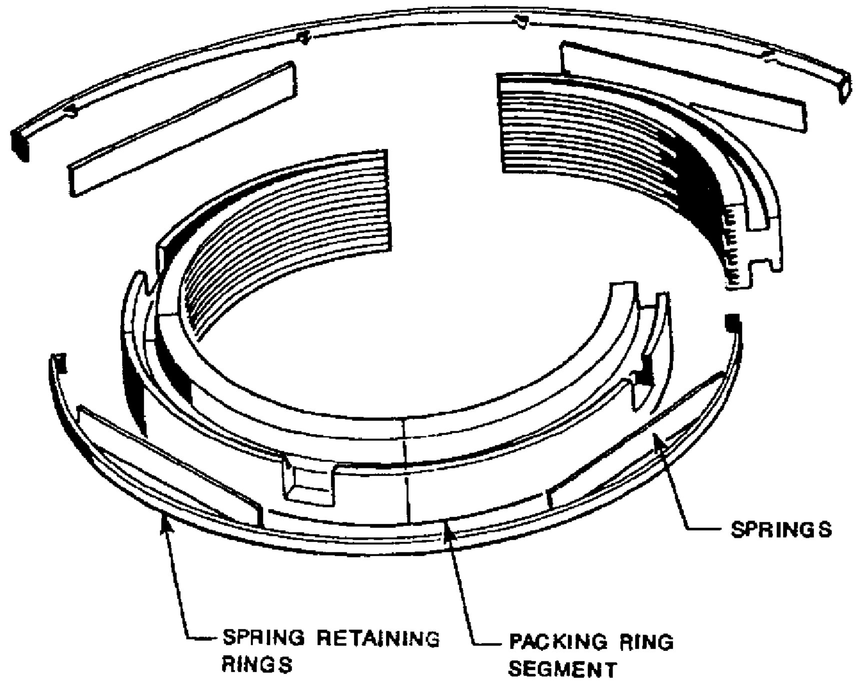


Figure 231-2-7 Shaft Packing

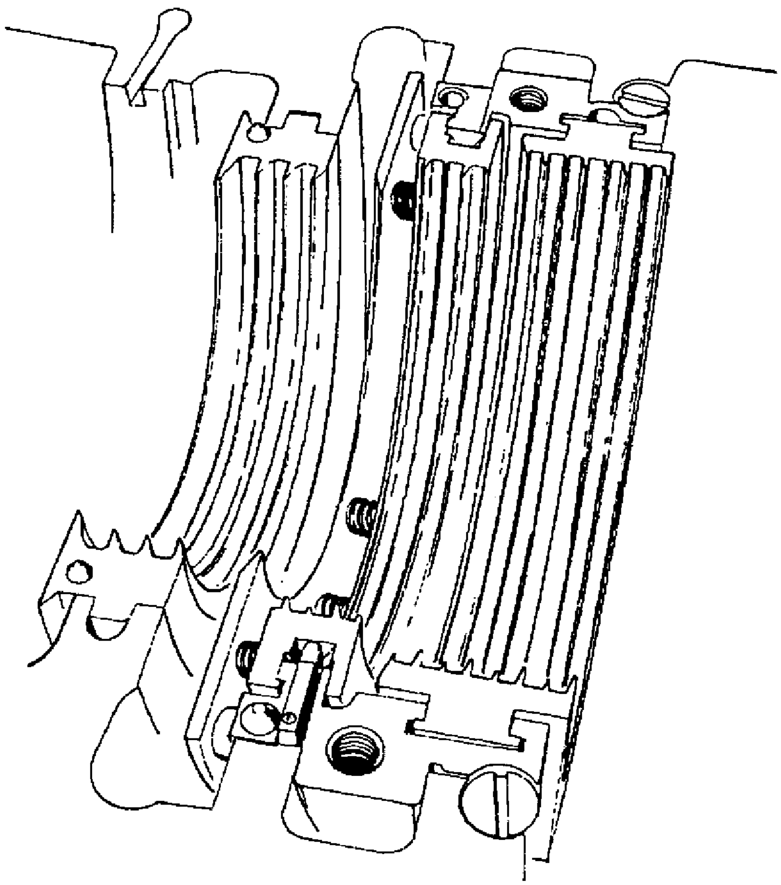


Figure 231-2-8 Gland-Packing Ring Assembly

231-2.5 DUMMY PISTON

231-2.5.1 Dummy pistons are provided only in older (Parsons-type) reaction turbines, such as AD 15 to 19 and AR 5 to 8, and are no longer being built for naval propulsion use.

231-2.6 GLAND SEALING SYSTEM

231-2.6.1 GENERAL. Although labyrinth shaft packings or control valve lift rod bushings limit the steam or airflow to small quantities, they do not cut off flow completely. The gland seal system, which includes supply and leakoff sections, is designed to provide the positive control required to keep steam from leaking out to the engine space or air from leaking into the main condenser through the turbine casings. A small amount of steam leaking into the engine room or small amount of steam or air leaking into the turbine would not normally require emergency repairs or delay deployment of ships. On certain labyrinth designs a whistling sound is normal when pulling proper vacuum during operation. While no leakage is expected, it is technically acceptable if steam leakage into the engine room is that of a wisp or less, which does not endanger personnel, overload compartment air conditioning, or cause water in the lube oil system. If turbine vacuum and oxygen level in the condensate are satisfactory, steam air leaking into the turbine is permitted if it does not cause the rotor to break away and roll with the throttles shut. Also, steam is expected to be seen during maneuvering while auto or manual regulator functions are set. Troubleshooting for the cause of steam leakage should always begin with the gland seal and exhaust system since these are the only positive controls of leakage. Gland leakage problems are rare if the gland seal and dump valves are adjusted correctly, if there are no low spots or loop seals in the gland exhaust line to collect condensate, and if the gland exhauster is providing the required leakoff vacuum at the turbine glands. Sometimes a slight steam leak in a bearing vicinity causes the thermometer on the sight flow indicator to give a false indication of the bearing oil temperature because of gland leakage steam blowing on it. This should be corrected at the next availability, if feasible. Steam from lift rods when first starting is normal (due to water flashing). Wait until the steam subsides.

231-2.6.2 SEALING RINGS. Even though the number of rings at any gland increases in proportion to the steam pressure to be sealed, the system could be functional with three rings at each gland. For simplicity, a three-ring example is used in this discussion. The space between the outer two rings connects to the leakoff system, which is maintained at a vacuum of about 10 inches of water, while the space between the two inner rings is connected to a low-pressure (0.5 to 3.0 psig) gland seal system or steam seal system.

231-2.6.3 FLOW REQUIREMENTS. With these pressures fixed, air always flows inward through the outer ring and steam always flows across the middle ring, directed toward the leakoff space. The inner ring passes only steam, but flow may be in either direction, depending on casing internal pressure.

231-2.6.3.1 Pressure variations in the casing (from standby or astern operation through the range to full power) impose various gland seal system requirements. These requirements include supplying steam to the turbines at low power (a balanced condition where high-pressure glands supply steam only to low-pressure glands at some intermediate power), and dumping excess steam to the condenser at high powers.

231-2.6.4 AUTOMATION. The system is automated by pressure-regulating valves. One valve supplies steam when pressure is below a set point; another valve dumps steam when pressure is above the set point.

231-2.7 TURBINE DRAINS

231-2.7.1 GENERAL. Turbine casings are fitted with external and internal openings to drain water from low spots to drain systems or to the main condenser. Proper drainage at lightoff is important to dispose of water accumulated during shutdown and water produced when initially introduced steam condenses on the cold metal surfaces. Inadequate drainage could produce local chilling of the rotor, roughness in turbine startup, or damage from standing or solid water striking the rotor. Opening the drains during the securing procedure is important to drain and dry out turbines properly and to avoid corrosion of some steel parts during idle periods.

231-2.7.2 EXTERNAL CASING DRAINS. Superheated stages with external casing drains are fitted with manually operated valves that are fully opened during turbine warm-up and left partially open during initial low-speed operation. Normally, these valves can be closed after about 10 to 15 minutes of low-speed, underway operation since drainage should then be completed. Avoid leaving these drain valves cracked unnecessarily because turbine power loss results when steam blows through the drain line and bypasses turbine stages. All stages operating in the moisture region are fitted with orifices that continuously drain collected moisture to the condenser.

231-2.7.2.1 Drain lines containing steam traps automatically drain accumulated water, if functioning properly. These traps should be checked periodically to ensure that they are not blocked or that they do not blow through continuously. Either condition will adversely affect turbine operation. Steam trap types found in naval service are bimetallic, thermostatic, thermodynamic, and impulse.

a. Bimetallic Steam Trap. A bimetallic steam trap operates on the theory that dissimilar metals will expand differently when heated. When the fluid temperature acts on the bimetallic sensing element, the trap will close if steam is present or open if condensate is present. The bimetallic steam trap is used primarily for draining main and auxiliary steam lines.

b. Thermostatic Steam Trap. This trap is controlled by expansion of vapor from a volatile liquid in a bellows-type element. Its use is limited to pressures up to 100 psi. These traps are used primarily in the constant service steam system and auxiliary exhaust system.

c. Thermodynamic Steam Trap. The only moving part in this trap is a hardened stainless steel disk. Condensate flows from the trap when the pressure of the condensate or air lifts the disk off its seating surface. The condensate flow continues until the flashing condensate approaches steam temperature. The high-velocity jet of flash steam reduces the pressure under the disk, while the pressure above the disk increases, closing the disk. As the trapped steam condenses, pressure above the disk decreases. The disk is then lifted by the inlet pressure and condensate is discharged. The thermodynamic steam trap is used for draining main and auxiliary steam lines.

d. Impulse Steam Trap. The impulse steam trap operates on the fact that hot water under pressure will flash into steam when the pressure is reduced. The only moving part in this trap is the disk. The disk is an unusual design. Near the top of the disk is a flange. The working surface above the flange is larger than that below the flange. A control orifice runs through the disk. The top of the orifice is smaller than the bottom. The differences in area above and below the disk and of the orifice through the disk cause the trap to open or close depending on the fluid pressure and temperature. This trap can be used in pressures up to 2,500 psi and temperatures of 1050°F.

231-2.7.2.2 For further detail on steam traps refer to NAVSEA 0910-LP-106-1000, Volume I, Maintenance Manual for Valves, Traps, and Orifices (Nonnuclear), User’s Guide and General Information, and NSTM Chapter 505, Piping Systems .

231-2.7.3 INTERNAL DRAINS. Where it is impractical to drain turbine low spots externally, particularly where the stage operates in the moisture region, fixed internal drains are provided to drain water that would otherwise be trapped. Stage-to-stage drainage may be accomplished by connecting the drains of separate stages or by weeping the flow from stage to stage through orifices, openings, or cutaway drain blades. The hole sizes are a compromise: they must be large enough to ensure drainage but small enough to minimize leakage. The smaller sizes are susceptible to plugging and should be checked (if and when access is available) to ensure that they are clear. Drain orifices, in low-pressure turbines ahead and astern element casings, drain directly to the condenser and, in larger turbines, are accessible by entering the turbine. Internal drains in high-pressure or single-casing turbines and in turbine gland areas are normally accessible only with disassembly. Study the technical manual or detailed drawings to locate all drains and to establish a feasible routine, or as specified in the PMS, that will ensure inspections. Schedule internal drain inspections during overhauls when the turbine casing is to be opened. The inspection is to determine if the drain is plugged, evidence of erosion and corrosion, wall thickness and integrity, and drain opening size.

231-2.7.4 OIL-WATER DRAINS (SLOP DRAINS). The slop drain collects soil and water that has leaked from the oil seal ring and gland seal ring assemblies, respectively. The slop drain is open to the atmosphere and must be checked for free flow so that lube oil and water does not leak across the shaft. If the drain is plugged, lube oil can leak across the shaft and cause the lube oil to be baked and carbonized near the seal ring. Carbonized lube oil can cut the shaft. Water that has leaked across the shaft can enter the lube oil system causing a reduction in the lube oil lubricating properties. Slop drain inspection should be included in the maintenance schedule to determine if the drain is free flowing to prevent damage to the shaft and lube oil systems.

231-2.8 OIL DEFLECTORS

231-2.8.1 Oil deflectors are also known as oil seals and oil seal rings. Oil deflectors keep oil from leaking out of the turbine bearing brackets. Propulsion and SSTG turbine oil deflectors are of the labyrinth (ring) type, located adjacent to, but in a rotor section of larger diameter than, the bearing journal. Most of the oil leaving the oil deflector side of the journal bearing is diverted to drain by the centrifugal action of the shaft shoulder. The deflector, because of its small clearance with the shaft, limits the oil vapor or oil tending to flow along the shaft from windage pressure developed on the bearing side of the deflector. The exit side of the deflector is open to atmosphere in the top half through the space between the bearing cap and the turbine housing. This opening maintains atmospheric pressure and permits gravity draining of the slop drain chamber between the gland and oil deflector rings. The mixture of oil and water collected in this chamber drains directly from a bottom opening to the outside or drains indirectly through connected piping.

231-2.9 THRUST BEARINGS

231-2.9.1 GENERAL. Each main propulsion and SSTG turbine has a thrust bearing that performs two important functions. It establishes the axial position of the rotor within the close axial clearances. It also sustains the axial forces, in either direction, imposed on the rotor from the component weight of the raked shaft, unbalanced steam forces, and forces developed in sliding the flexible coupling.

231-2.9.1.1 Most thrust bearings are of the Kingsbury pivoted-segmental-shoe type, but some low-force applications use nonsegmental or tapered-land bearings. The pivoted-shoe-type bearing is by far the most common and is described in detail in paragraphs 231-2.9.2 through 231-2.9.2.9.

231-2.9.2 KINGSBURY PIVOTED-SEGMENTAL-SHOE THRUST BEARING CONSTRUCTION. A pivoted-shoe thrust bearing (Figure 231-2-9) consists of seven basic parts and functions as described in paragraphs 231-2.9.2.1 through 231-2.9.2.9. After replacing any of these parts, adjust thrust bearing oil clearance and rotor axial position as described in paragraphs 231-6.3.2 and 231-6.3.3.

231-2.9.2.1 Collar. The collar is secured to (or is integral with) the shaft and rotates with the shaft. Each collar face is a finely machined or ground rotor positioning surface and is the surface through which thrust force is transmitted. Maintain the collar face flatness, parallelism, and surface finish at drawing requirements. A collar on the shaft with half of the shoes and leveling plates is shown in Figure 231-2-10.

231-2.9.2.2 Shoes. Each shoe is babbitt-faced on the collar side with a hardened button on the opposite side that contacts and is supported by one element of the leveling plate assembly. Each shoe is free to tilt and form the hydrodynamic wedge between its babbitt-face and the collar. Although noninstrumented shoes may be interchanged, retaining original shoe position is good engineering practice. Shoes may be replaced one at a time, once properly measured.

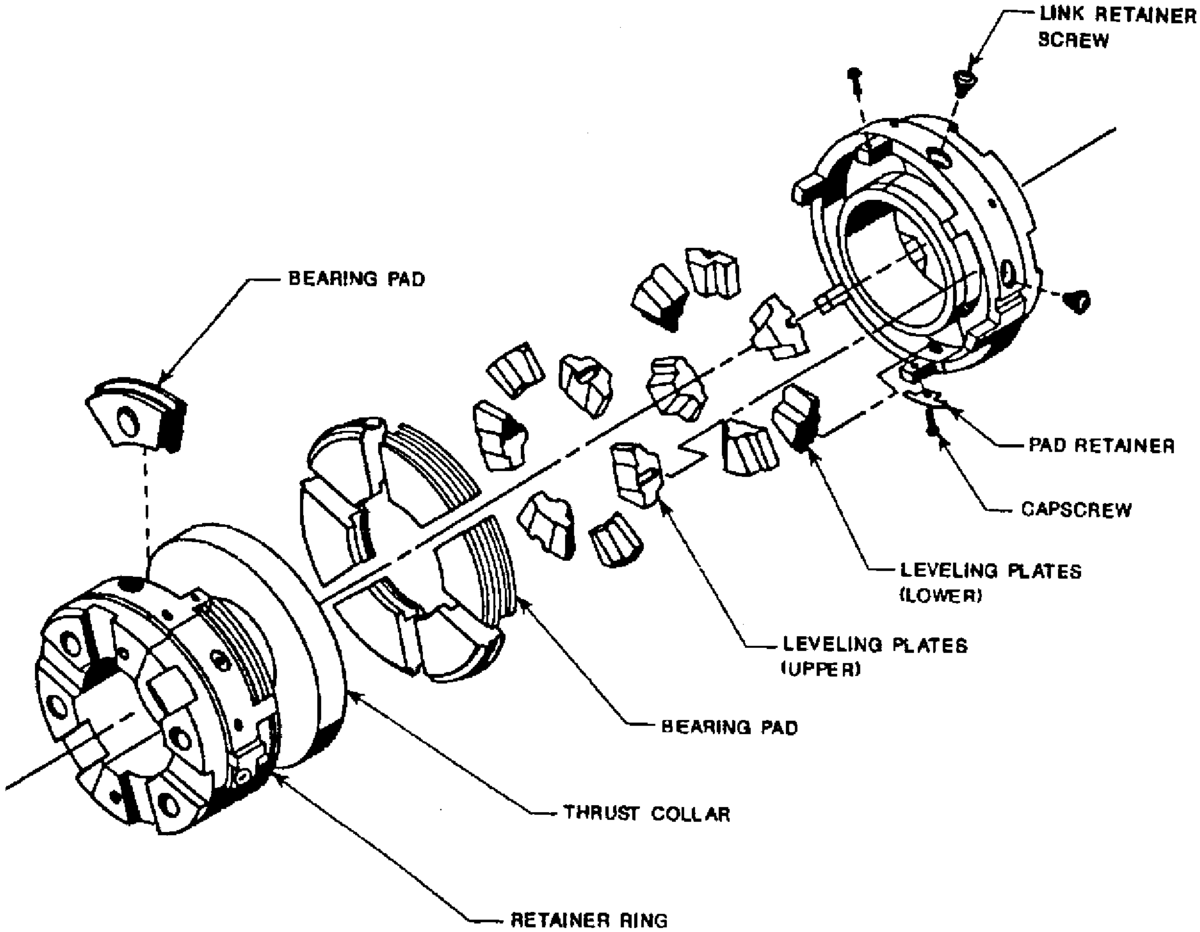


Figure 231-2-9 Thrust Bearing (Exploded View)

231-2.9.2.3 Leveling Plates. Leveling plates are located between the shoes and the base ring. The plates provide point support for shoe and line contact with the base ring, permitting tilting of the shoe and rocking of leveling plates necessary to distribute thrust load equally on each shoe.

231-2.9.2.4 Base Ring. The base ring maintains the alinement of parts and transmits thrust force to the bearing housing.

231-2.9.2.5 Oil-Control Ring. The oil-control ring encloses the thrust collar and film area, and collects and controls the quantity of oil flowing through the thrust bearing.

231-2.9.2.6 Oil Seal. Oil seals are normally close-clearance brass rings enclosing the shaft at each end of the thrust bearing. Oil seals limit oil leakage out of the thrust bearing, permitting flooded operation.

231-2.9.2.7 Spacers or Filler Pieces. Spacers, or filler pieces, are rings used in thrust bearing assemblies to set thrust clearance (one ring) and rotor position (two rings).

231-2.9.2.8 Tilt Angle. In thrust bearing operation the rotation of the thrust collar forces oil under one end of each shoe, causing the shoe to tilt and support the thrust load (somewhat like a water skier's weight is supported by action of the water on tilted, moving skis). Increasing speed normally increases tilt angle, and increasing the load decreases the angle. The edge of each shoe should have a noticeable chamfer (typically 1/16 inch). A shoe with no chamfer may not form an oil wedge and may run hot and wear prematurely.

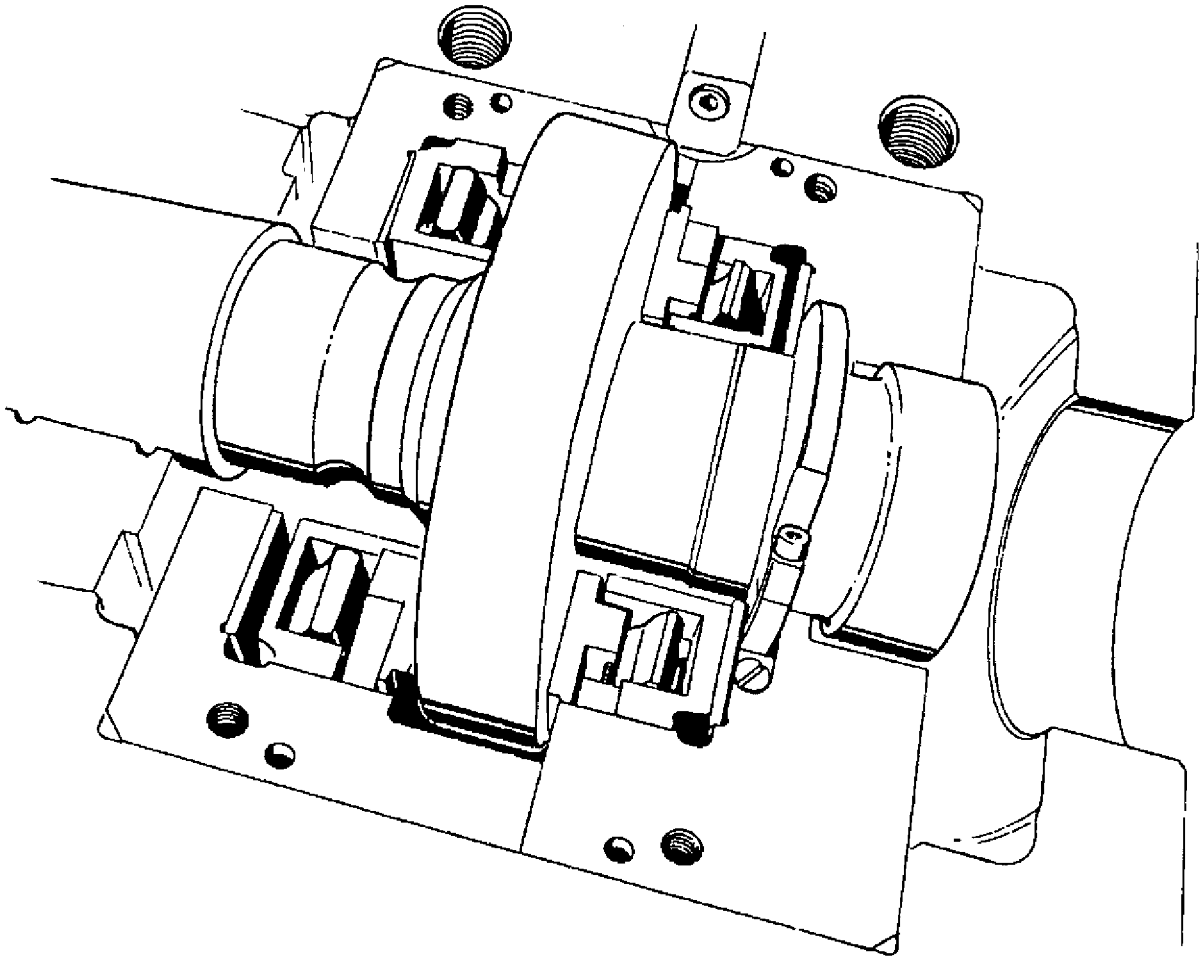


Figure 231-2-10 Thrust Bearing Collar

231-2.9.2.9 Misalinement. The leveling action of the Kingsbury bearing makes it less prone to misalinement than bearings of other types. Minimizing collar runout and precisely manufacturing spacer pieces, when required, is conducive to successful operation at lower temperatures and tends to minimize wear.

231-2.9.3 NONSEGMENTAL THRUST BEARINGS. The nonsegmental thrust bearing consists of a thrust collar keyed and secured to the shaft by locknuts or other devices and two babbitt-faced thrust plates between which the collar rotates. In some turbines one end of a journal bearing is faced with babbitt and serves as a thrust plate. In still others two collars are secured on the shaft, one at each end of a journal bearing; and each end of the journal bearing has a vertical babbitted face that serves as a thrust plate. Some older turbines manufactured by the General Electric Company for driving generators were equipped with nonsegmental thrust bearings known as tapered-land thrust bearings. In this type of thrust bearing, the stationary surface on the side that absorbs the thrust is a solid ring with a babbitted surface. A series of about six radial grooves are cut in the babbitt to distribute oil. Each babbitted surface between the radial oil grooves is machine tapered in both the circumferential and the radial directions to provide a wedge-shaped oil film. The inactive stationary thrust surface is not tapered. With proper lubrication the thrust plate lands should experience little or no wear, and the designed end play will change very little with service. Shims are generally provided between the thrust plates and their housings. End play adjustment, if necessary, can be accomplished within reasonable limits by changing the thickness of the shims. In later tapered-land bearings the entire bearing pivots on a yoke and holding plate located at the end of the bearing assembly. When dismantling nonsegmental thrust bearings for inspection, carefully clean the grooves in the thrust plates that distribute oil over the thrust surfaces to ensure a free flow of oil.

231-2.10 JOURNAL BEARINGS

231-2.10.1 DESCRIPTION. Main turbine journal bearings are small clearance-type bearings that operate on a hydrodynamic film principle. For hydrodynamic lubrication to exist, an adequate supply of lubricant is required, which may or may not be pressurized. In hydrodynamic lubrication the load-carrying surfaces of the bearing are separated by a thick film of lubricant, which prevents metal-to-metal contact. The film pressure is created by the rotating journal pulling the lubricant into a wedge-shaped area (oil wedge) at a velocity sufficiently high to create the pressure necessary to separate the surfaces against the load on the bearing. Functionally, the bearings maintain the rotor in such a radial position that will avoid rubbing between rotor and casing elements. The bearings are statically loaded by the weight of the rotor and by any radial asymmetrical steam forces that may exist. The bearings may also support dynamic loads from mechanical rotor imbalance and coupling force.

231-2.10.2 BEARING CLEARANCE. Equipment technical manuals or detail drawings will normally list minimum and maximum design bearing clearances. When unavailable use the rule of thumb that the nominal bearing clearance in mils equals 1-1/2 times the journal diameter in inches.

231-2.10.3 JOURNAL BEARING TYPES. Journal bearings have oil-lubricated sliding surfaces of one of the basic types described in paragraphs 231-2.10.3.1 through 231-2.10.3.9.2 and illustrated in Figure 231-2-11.

231-2.10.3.1 Plain Cylindrical (Sleeve) Bearings. Each bearing consists of two 180-degree semicylindrical shells that are babbitted and cylindrically bored. No grooves are machined in the babbitted surfaces other than axial oil distribution grooves at the oil inlets and circumferential oil drain grooves at one or both ends of the bearing (Figure 231-2-11, view A).

231-2.10.3.2 Top-Relieved or Overshot Bearings. Top-relieved or overshot bearings are similar to plain cylindrical bearings except that bearing metal is removed from a relatively wide circumferential groove in the center of the upper, or nonloaded, half of the bearing shell to reduce bearing power loss (Figure 231-2-11 , view B).

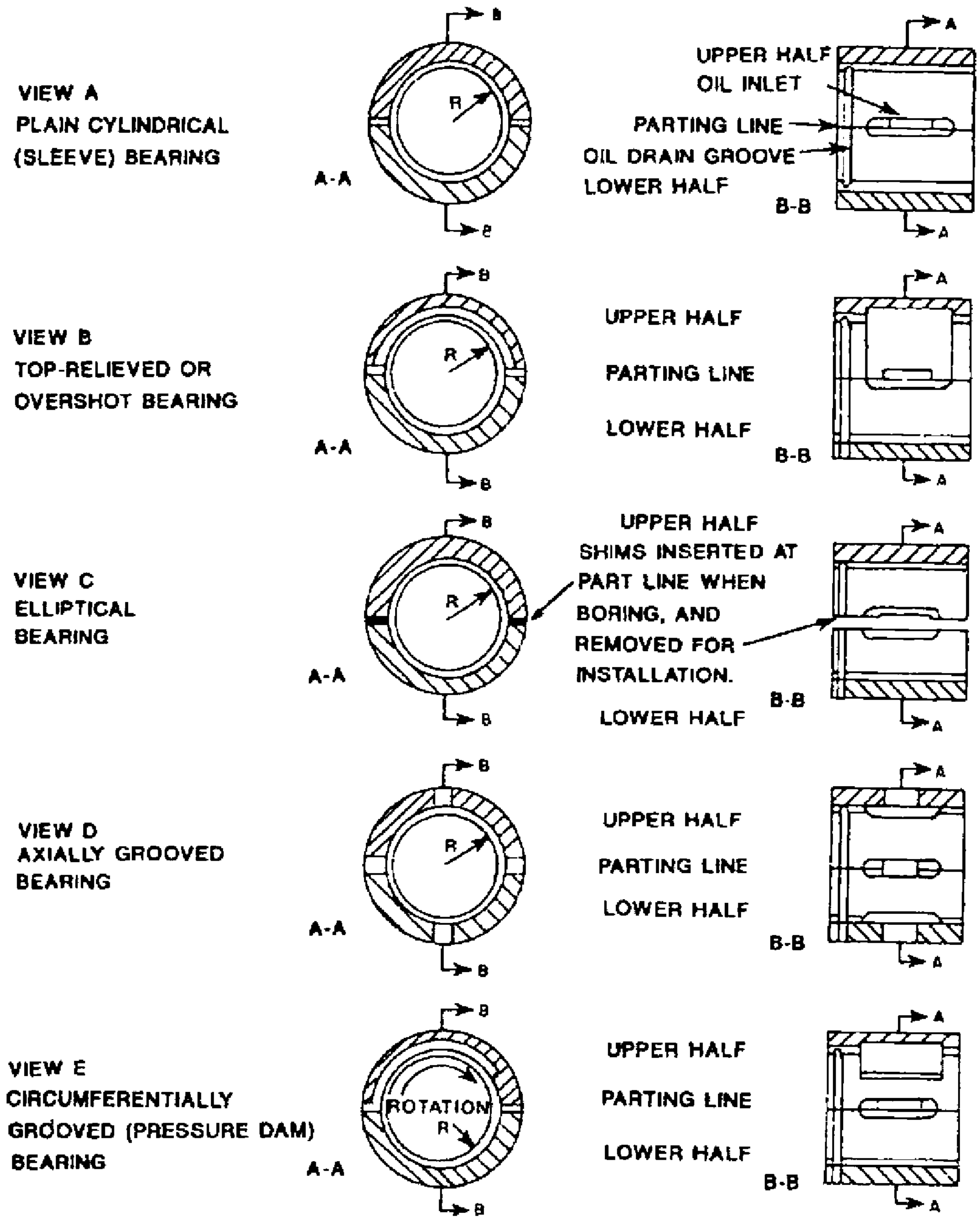
231-2.10.3.3 Elliptical Bearings. Elliptical bearings are also similar to cylindrical bearings except that bearing halves, when bored, are separated with shims at the parting lines. To give the elliptical configuration, these shims are removed from the bearing when installed in the turbine (Figure 231-2-11, view C). The elliptical opening, in this type bearing, is generated by cylindrically boring the bearing with an appropriate shim at the bearing parting face. Radial clearance at the horizontal joint (cheeks) is then 1/2 the shim thickness greater than the top or bottom clearance. Elliptical openings tend to improve oil feed to the loaded lower half of the bearing.

231-2.10.3.4 Thin-Shell Bearings. Elliptical bearings are most often seen in thin-shell bearings where the added side clearance will tend to counteract a reduction in this clearance from the movement of the free ends toward the journal. The thin-shell bearing is characterized by a shell-thickness-to-inside-diameter ratio of 0.125 or less, or a shell outside-diameter-to-bearing-inside-diameter (OD/ID) of 1.25 or less. Thin-shell bearings are seen in gear train bearings and some older turbines.

231-2.10.3.5 Axially Grooved Bearings. Axially grooved bearings are similar to plain cylindrical bearings except that these bearings have four or more axial (longitudinal) oil inlet grooves located symmetrically around the upper and lower halves of the bearing. This improves the stability of the high-speed journal bearings against the effects of oil whip and oil-film whirl (Figure 231-2-12 and Figure 231-2-11, view D).

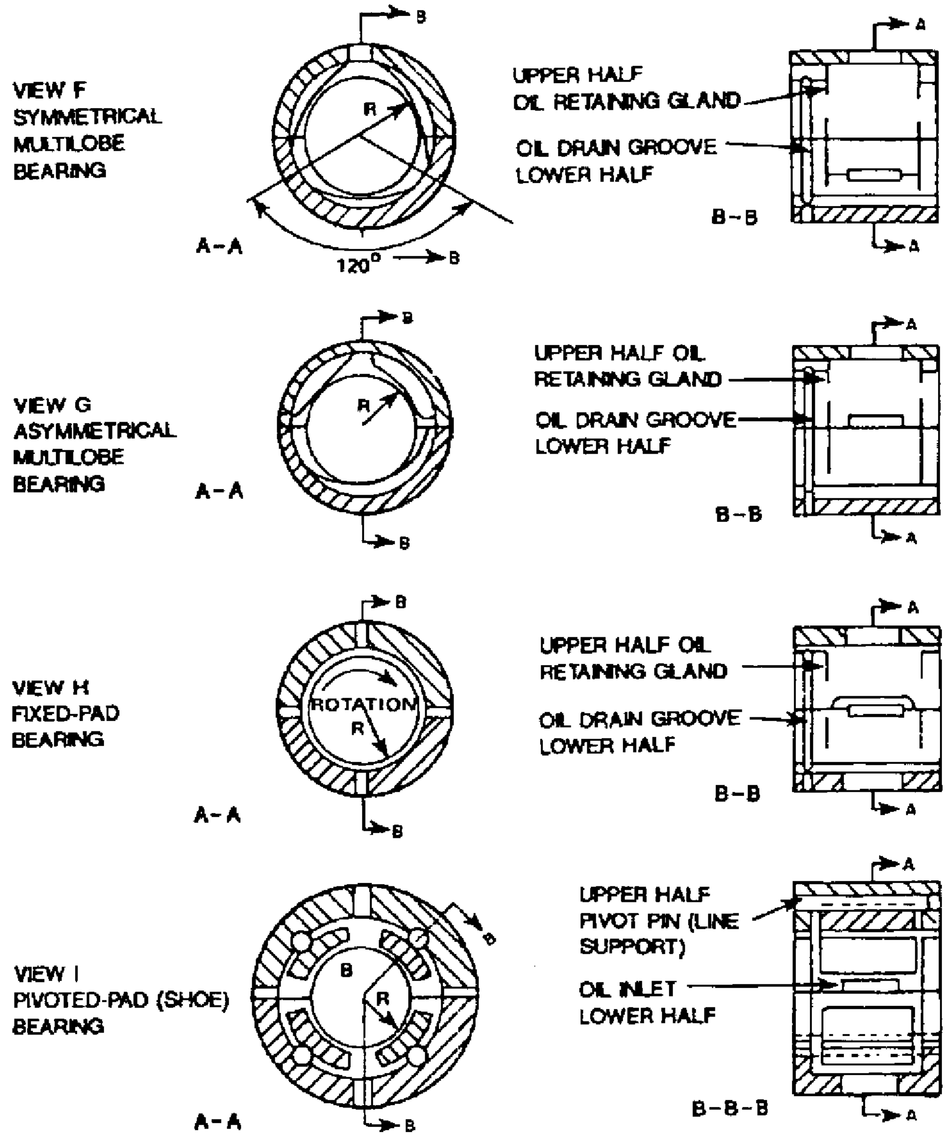
231-2.10.3.6 Circumferentially Grooved Bearings. Circumferentially grooved (pressure dam or pressure pad) bearings are similar to cylindrical bearings except that these bearings have a wide circumferential groove starting at the parting line in the upper or nonloaded shell in the direction of ahead shaft rotation and terminating in the upper shell at a sharp dam approximately 45 degrees beyond the vertical.

231-2.10.3.6.1 The dam traps oil above the journal and develops a hydrodynamic pressure that adds to the bearing load carried by the lower shell (Figure 231-2-11, view E).



NOTE: SELF-ALINING OR RIGID MOUNT OPTIONAL CYLINDRICAL BACK AND RIGID MOUNT SHOWN.

Figure 231-2-11 Sliding-Surface-Type Turbine Bearings (Sheet 1 of 2)



NOTE: SELF-ALINING OR RIGID MOUNT OPTIONAL. CYLINDRICAL BACK AND RIGID MOUNT SHOWN.

Figure 231-2-11 Sliding-Surface-Type Turbine Bearings (Sheet 2 of 2)

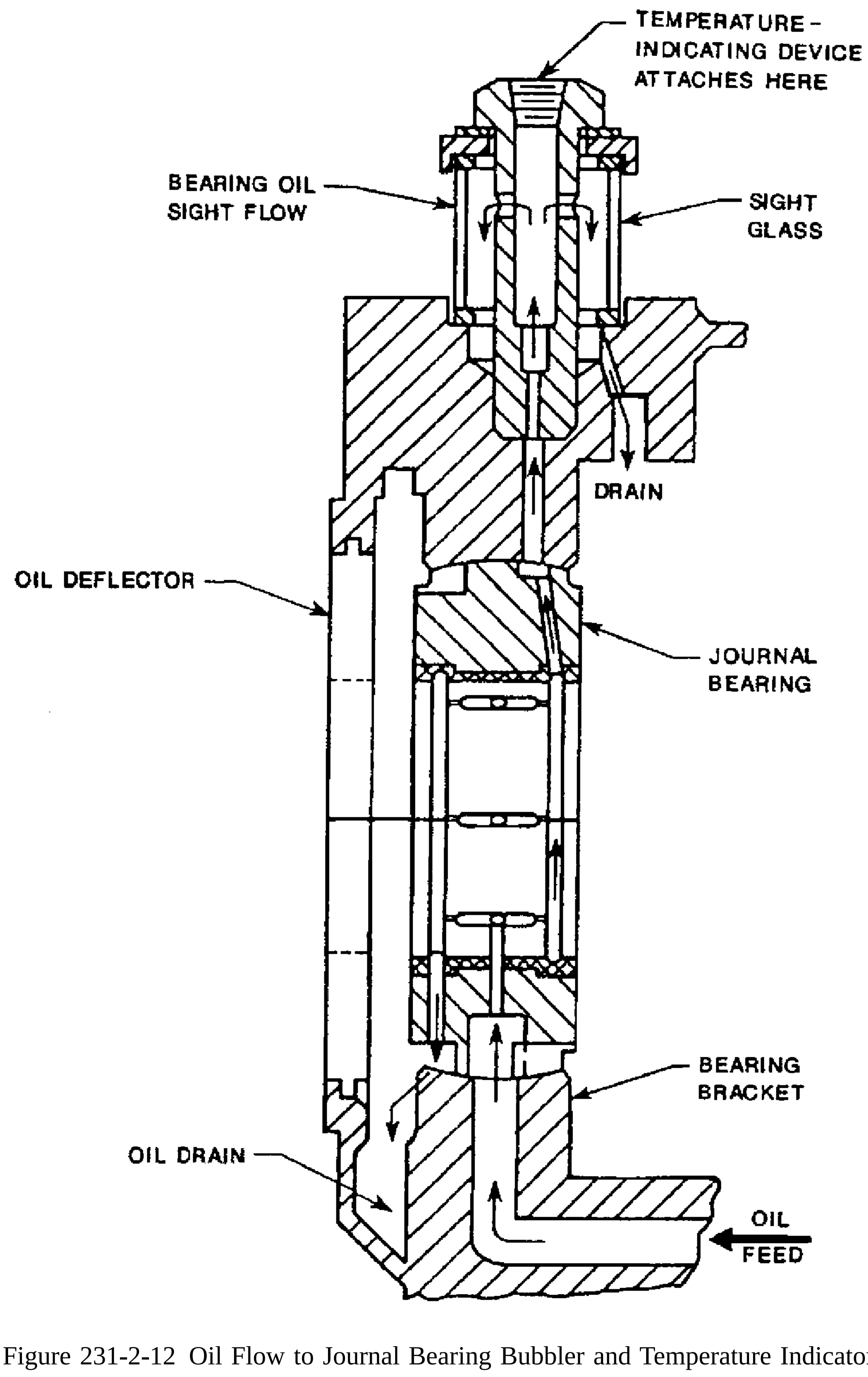


Figure 231-2-12 Oil Flow to Journal Bearing Bubbler and Temperature Indicator

231-2.10.3.7 Multilobe Bearings. Multilobe journal bearings are either symmetrical or asymmetrical (Figure 231-2-11, view F, and, view G, respectively), with load and speed capacity for shaft rotation in either direction. The most common multilobe bearing design is the three-lobe symmetrical bearing, consisting of three 120-degree arcs whose centers are displaced from the bearing center. The three-lobe asymmetrical bearing consists of one 180-degree arc in the lower half and two 90-degree arcs in the upper half, with centers displaced from the bearing center.

231-2.10.3.8 Fixed-Pad Bearings. Fixed-pad bearings have three or more circumferential arcs of bearing surface that taper or converge toward the journal in the direction of ahead rotation and then become concentric with the bearing center. Fixed-pad bearings function the same as the multilobe bearings except that fixed-pad bearings have an extremely limited capacity for shaft rotation in the reverse direction. They are used, therefore, for shafts that rotate in one direction only (Figure 231-2-11, view H).

231-2.10.3.9 Pivoted-Pad or Shoe Bearings. Pivoted-pad or shoe-type bearings have a minimum of four, and often five circumferential shoes. Each shoe may be pivoted by a pin or dowel parallel to the journal axis, or by a button. This design is considered to offer the most positive solution for eliminating shaft vibration caused by oil whip and oil-film whirl (Figure 231-2-11, view I). This bearing can carry loads equally well in all directions (and in either direction of rotation).

231-2.10.3.9.1 Pivoted-pad bearings operate on essentially the same principle as Kingsbury thrust bearings except:

a. The shoes act radially on the journal rather than axially on the thrust collar.

b. There are no leveling plates; therefore, shoes transmit forces directly to the bearing shell.

c. Forces are not equal on all shoes but are shared by more than one shoe in the direction of applied force.

231-2.10.3.9.2 To a practical extent, shoes are interchangeable within the bearing and between the fore-and-aft and port-and-starboard applications of the same design. Although parts are interchangeable, once a bearing has been used, it is good engineering practice to retain the shoe position and orientation. One shoe may be replaced at a time once the proper measurements have been taken.

231-2.11 SIGHT FLOW INDICATORS OR BUBBLERS

231-2.11.1 JOURNAL BEARING BUBBLERS. Sight flows, or bubblers, fitted to the bearing assembly permit ready verification of oil flow to the bearing. A journal bearing, showing oil flow to bearing bubbler and thermometer, is illustrated in Figure 231-2-12. Depending on design and operating speed, the oil flow as seen in the sight glass may vary in quantity from a weak indication to fully flooded and from clear oil to foamy oil without loss of lubricating properties. These are all normal indications, which usually require no special action.

231-2.11.2 BULLSEYE BUBBLERS. Bullseye bubblers are sight flow indicators for thrust bearings on many main turbine designs. A bullseye, or glass window, is recessed in the top of the bearing bracket housing, allowing the operator to look down at the oil flowing in the thrust bearings. At high speed, oil will be flung against the bullseye, giving a flooded appearance, and entrained air will give the oil a frothy, milky appearance, making flow indications unclear. These are normal expected conditions that should not restrict turbine operation. Clear flow indications will return at lower speeds.

231-2.11.3 For more information on sight flow indicators, see paragraph 231-6.4.7.

231-2.12 TYPES OF TURBINE BLADES

231-2.12.1 TEE-ROOT TYPE. These are for installation in circumferential grooves. A tee-root is shown in Figure 231-2-13, view A. This root is not tight in the groove until the caulking or locking piece is installed. There may be a partial clamping on the side lugs (or straddles) after assembly, depending on the initial clearance and amount of caulking. On rotation, the centrifugal force tends to spread the groove walls and increase the clamping at the straddles. Another version of this type of root is the double-tee, which has greater strength to resist centrifugal forces and is used for larger blades. Generally, when the tee- or double tee-root is to be mounted on a disk, the straddle design is used. When the blades are for installation on a drum-type rotor, the straddle design is not used.

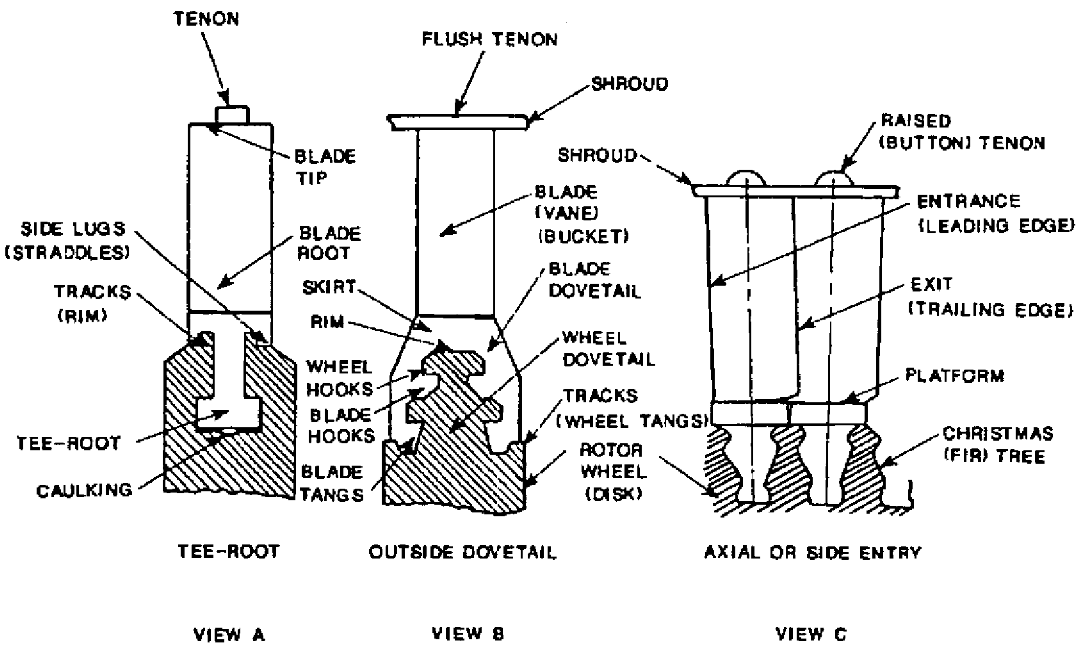


Figure 231-2-13 Tee, Outside-Dovetail, and Axial-Entry Blades

231-2.12.2 OUTSIDE-DOVETAIL TYPE. This type, is shown in Figure 231-2-13, view B, is designed for circumferential entry. On assembly it should have a light drive fit from the entry slot to the installed position. The disk has straddle grooves that lock the blade dovetail axially and prevent spreading caused by centrifugal force. This type of blade may have single, double, or triple tangs.

231-2.12.3 AXIAL-ENTRY TYPE. This root, shown in Figure 231-2-13, view C, is driven into axial slots broached or milled in the disk rim. It is usually secured by some type of platform interlock, pin, or key arrangement. A thinner disk can be used with this root than is possible with any other root design. In case of damage requiring blade replacement, it is possible to replace only the blade group or groups containing defective blades. Another axial entry design not shown in Figure 231-2-13, view C, is the ball and shank design, which may be either equal length shank or a long or short configuration, depending on wheel diameter.

231-2.13 SUPPORTS

231-2.13.1 Steam turbines are supported structurally by many types of foundations, including the ship's foundation, a bedplate (or subbase), a box girder, the main condenser, or the gear case. Typically, one end of the turbine is supported but not restrained, and keys are used on that end, instead of studs or bolts, to allow for thermal growth of the casing. Chocks are specially machined shims used to aline the turbine and equipment during installation. On surface ships, propulsion or SSTG equipment is hard-mounted directly to the ship's structural members. On submarines, the main propulsion unit or complex (MPU or MPC) is manufactured with a subbase or bedplate that is the structural foundation supporting the propulsion turbines, propulsion gears, and associated equipment. This subbase is then soft-mounted to the ship's structure with an arrangement of shock mounts and snubbers. The SSTG on submarines includes the turbines, generators, and associated equipment mounted on a subbase that is soft-mounted to the hull. Check all structural fasteners periodically. Loose fasteners can cause equipment noise and vibration. Do not attach unauthorized equipment to a soft-mounted subbase.

231-2.14 LAGGING

231-2.14.1 Lagging is the thermal insulation attached to the turbine with hooks or studs. On non-carbon-steel casings hooks or studs are welded on only before heat treating. Do not operate turbines for any great length of time with any lagging removed as lagging is important to the heat distribution of the casing. Asbestos lagging should be replaced with nonasbestos lagging only by a qualified facility. Protect or shield lagged areas that see traffic or might see oil leakage.

231-2.15 TURNING OR JACKING GEARS

231-2.15.1 GENERAL. Most main propulsion complexes with propulsion steam turbines and most SSTG steam turbines are supplied with turning or jacking gears to rotate the rotors at slow speeds. For more information on propulsion turning gears, refer to NSTM Chapter 241, Propulsion Reduction Gears, Couplings, Clutches, and Associated Components .

231-2.15.2 DESIGNS. Turning or jacking gears are driven by an electric motor or a hydraulic motor. They usually rotate an attached gear that is engaged to mesh with a gear on the SSTG rotor or propulsion gear element to be driven. Some are ratchet style, moving a toothed bar up and down to rotate the driven rotor in small, intermittent steps. Most steam turbines have a hex head nut on the end of the rotor so that it can be manually turned or barred over when necessary. Some earlier ships had low-pressure steam fed directly into the propulsion turbine or directly into a small ″piggyback″ turbine for low-speed turning, but these designs have been abandoned.

231-2.15.3 PURPOSES. Turning or jacking gears are used to do the following:

a. Warm up steam turbine rotors with gland seal steam on the glands on the ends of the turbine rotors to allow even distribution of heat along the rotor. This will help reduce thermal shock and possible thermal stresses when steam is first admitted through the throttle valves into the turbine. See paragraph 231-3.2.2 for more information.

b. Cool down steam turbine rotors to allow an even distribution of heat along the rotor and to help reduce thermal stresses.

c. Prevent bowing of the turbine rotor by evenly distributing heat from turbine operation, heat from the turbine glands, and the coolness of the condenser. This is especially true after operation in superheated steam.

d. Allow slow rotation of the rotor or gear train for inspections.

e. Allow positioning of the rotor or gear train for maintenance. See paragraph 231-3.2.4.

SECTION 3

STEAM TURBINE OPERATION

231-3.1 OPERATING INFORMATION

231-3.1.1 GENERAL. There are basic similarities in the principal types of turbine propulsion systems and even greater similarities in the various designs of each type. There remain, however, many detail differences in individual plants that make it impossible to present in this section a complete operating procedure for main propulsion and ship service turbine generator (SSTG) turbines. It is necessary for the ship's force to establish the routines necessary for operating specific plants.

231-3.1.2 GENERAL OPERATING PROCEDURES. The general operating procedures for main propulsion and SSTG turbines is given in the manufacturer's technical manual and applicable ship's documents. Generally, these manuals and documents will specify the following:

a. Ensuring that turbine is free to rotate and free of all loose material
b. Maintaining lube oil temperature at 90°F minimum
c. Maintaining condenser vacuum
d. Ensuring that overspeed limiters, trips, and all other safety devices are functioning properly
e. Ensuring that the gland sealing steam system is operating
f. Ensuring that all drain valves are opened
g. Putting the auxiliary condenser into service

231-3.2 PREPARATIONS FOR GETTING UNDERWAY

231-3.2.1 PROCEDURE. Each ship should have a detailed procedure for lighting off the main propulsion and SSTG turbines. The manufacturer's technical manual devotes a chapter to operation and should be used in preparing the procedure. The propulsion turbine and SSTG manual specifies turbine procedures in detail but normally only briefly discusses support systems (those systems not provided with turbines) as they affect the turbine. Any overall step-by-step procedure shall combine procedures in lighting off and operating these support systems. The procedures shall include a checkout of any special monitoring instruments and establish which operations are to be reported to (or cleared by) higher authority (see paragraph 231-3.2.3 for precautions when turning shaft).

231-3.2.2 REASONS FOR PROPER WARM-UP. Warm-up procedures normally specify:

a. Establishing lube oil system pressure and heating lube oil to a minimum of 90°F. Maintaining steam turbine lube oil (2190 TEP) pressure and temperature during operation is vital to ensuring that the viscosity of the lube oil is within design. The viscosity of an oil is its tendency to resist flow or change of shape. A liquid of high viscosity flows very slowly. The higher the temperature of an oil, the lower its viscosity becomes. Lowering the temperature increases the viscosity. Also, viscosity requirements vary directly with load and the magnitude of clearances and tolerances. A higher viscosity oil can cause excessive turning resistance during cold starting. If the lube oil temperature becomes excessive, the drop in viscosity could cause high wear or even seizure. The viscosity must always be high enough to keep a good oil film between moving parts; otherwise, there will be increased friction, power loss, and rapid wear on the parts. Maintaining oil temperature to the unit between 120°F and 130°F provides optimum viscosity.

b. Turning turbine rotors by jacking gear until just before steam is admitted to turbine blade path

c. Maintaining degraded vacuum

d. Spinning engine, alternately ahead and astern, every few minutes

231-3.2.2.1 Establishing Proper Operating Temperatures. By following these procedures and holding conditions for the times specified, turbine metal temperatures can be uniformly brought as close to operating temperatures as heat sources (lube oil, gland seal steam, and small quantities of main steam) will allow. On startup, improper warm-up can cause excessive thermal stresses, rotor-to-casing rubs from differential expansion, or excessive vibration from a bowed rotor or a hogging (humped upward) or sagging casing.

231-3.2.2.2 Differential Expansion. The simple fact that turbine materials expand when heated has a tremendous influence on turbine design. Most of the important clearances in the turbine are specified in thousandths-of-an-inch and are established from considerations of steady-state and expected transient differences in temperatures of turbine parts. By their very nature and principle of operation, the various parts of the turbine casing and rotor experience progressively lower temperatures going from inlet to exhaust as steam energy is converted into mechanical work. Axial temperature gradients, therefore, cannot be avoided. Fortunately, under steady thermal equilibrium, the steam temperature at a stage heats both the rotor and the casing. Radial and axial clearances will not be substantially different from cold clearances. On the other hand, the casings and rotors have unequal masses and different heat transfer areas and coefficients, and this causes different heating rates under transient conditions. Rapid and large changes in temperature will therefore produce large changes in clearance. Conversely, warming the turbine properly will minimize temperature and clearance changes.

231-3.2.2.3 Unequal Heating and Thermal Stresses. Unequal heating is significant because a temperature gradient (variation in temperature over the length or breadth of part) will produce thermal stresses that are proportional to the steepness of the gradient. If high enough, such stresses can permanently distort or fracture the part.

231-3.2.2.3.1 This phenomenon can be illustrated by example. If a steel bar is heated from one side and kept cool on the other, the temperature difference between the heated and cold sides will induce stresses that bend the bar end towards the cold side. The temperature gradient produces distortion because the hot side tends to grow longer than the cold side. Since neither side is free to attain its equilibrium length, the heated side tends to stretch and therefore stresses the cold side. Similarly, the cold side tends to shorten the hot side and causes it to compress. If such stresses do not pass the yield point of the material, the bar is subjected only to elastic deformation and will return to its original size and shape when cooled to the starting temperature. Larger temperature gradients can be imposed, however, that will produce plastic or inelastic deformation and result in permanent distortion.

231-3.2.2.3.2 Thermal stress is so important that when permanent distortion is found in a turbine casing it would be reasonable to assume that the permissible temperature gradients were exceeded. One incident of an excessive gradient can produce distortion. A cyclic excessive gradient will most likely cause the part to fatigue crack or fracture.

231-3.2.2.4 Bowed Rotor. A turbine rotor can bow, or set, any time it sits idle in a hot casing. In a bowed rotor the rotor axial centerline is arched with respect to its cold or straight position. To avoid a permanent bow the rotor must be periodically rotated 180 degrees (5- to 8-minute intervals) unless specified in the manufacturer's equipment manual. Normally, the turbine would be put on turning gear and not left idle.

231-3.2.2.4.1 There is no way of directly measuring bowing in naval turbines. It can be recognized, however, by vibration at turbine rotational frequency occurring on startup following an idle rotor period. Vibration can be gradually reduced by rotating the turbine under heat for an extended period at low speed or a speed below that at which vibration occurs.

231-3.2.2.4.2 Bowing is caused by a temperature gradient across the rotor (as opposed to a gradient along its length), which causes a bend as described in paragraphs 231-3.2.2.3 through 231-3.2.2.3.2. The long, low-pressure (LP) turbine rotors are particularly susceptible to bowing since a given gradient will produce a rotor center deflection that is proportional to length. Also, the main condenser under the LP turbine tends to keep the lower half of the casing cool with respect to the upper half, setting up conditions that will produce higher temperatures in the upper part of the rotor. The gradient causes the rotor to bend upward. Similar, but less steep temperature gradients, exist in high-pressure (HP) turbines, where bowing also occurs.

231-3.2.2.4.3 Turning the rotors, either by jacking or alternately spinning rotors in the ahead and astern directions, avoids the temperature gradients and therefore the bowing. Spinning ahead is most effective in heating up the HP turbine, while spinning astern is most effective in heating up the LP turbine.

231-3.2.2.4.4 Unless the equipment technical manual specifies which valve (ahead or astern) to open first during alternate spinning (warm-up), the Engineering Officer should select and use the cycle that produces the best results in the propulsion plant. Because of the larger clearances that normally exist in the astern turbine, it is good practice to open the astern valve first, in case there is water in the steam line. Long double-flow LP turbines will be more sensitive to bowing because of their length and proximity to the cold condenser, and astern steam should normally be used to provide uniform heat to the rotor. For single-casing turbines, where all blading is on one rotor, it will make little difference which valve is opened first, provided ahead and astern steam are alternated to avoid putting way on the ship.

231-3.2.2.4.5 Heating of both turbines can be accelerated by warming up and initially operating at low main condenser vacuum (10 to 15 inches Hg).

231-3.2.2.4.6 For straightening a bowed rotor, see paragraphs 231-3.11 through 231-3.11.2.6.

231-3.2.2.5 Casing Distortion. The casing can be distorted when subjected to thermal gradients, despite stay rods and stiffeners (ribs, pipes, webs) placed in the casings to provide rigidity. Again, evenly and uniformly applying heat will relieve this condition. A hogging, or sagging, casing does not result in dynamic imbalance and vibration (as with a bowed rotor), but it can cause closure of small clearances (axial or radial) and cause a rub in the turbine. A distorted casing can also misaline the rotor and the attached gear and couplings, and increase wear on these components.

231-3.2.3 PRECAUTIONS WHEN TURNING SHAFT. The forced-lubrication system is usually operating before the rotors are turned over. Open steam line, strainer, throttle, and turbine drains to ensure that all water will drain from these parts. The main shaft bearing is essential to plant operation and should be checked for proper lubrication (see NSTM Chapter 244, Propulsion Bearings and Seals). Also, ensure that all personnel and foreign material are clear of the propeller shaft and propeller before turning over the shaft. Ensure that the inflatable shaft boot is deflated, if applicable. Ensure that the ship is properly moored and personnel are clear of mooring lines in case they part. Obtain permission from the Duty Officer before turning the shaft.

231-3.2.4 LOW-SPEED OPERATION WITHOUT FORCED LUBRICATION. Operate the lube oil (LO) system whenever the turbine is rotated, whether normal high-speed operation or low-speed rotation by turning gear, by jacking gear, or by hand. Turbines can be rotated under special circumstances at low speeds for short periods, however, without lube oil flow and without damaging bearing or journal surfaces.

a. If a turbine is discovered to be on turning gear without lube oil flow, take the following action:
   1 Stop and verify depth micrometer reading, if feasible.
   2 Verify that turning gear was not damaged while running without LO.
   3 Start the LO system.
   4 Verify proper flow at sight flow indicators.
   5 Verify that there is no babbitt in LO strainers.
   6 On subsequent operation verify proper temperatures and no resistance temperature element (RTE) alarms.
b. If an SSTG set must be rotated and lube oil flow is unavailable, normally because of LO system work, take the following action:
   1 Verify residual oil in bearing cavity and if necessary, add oil through sight flow indicator (SFI) openings.
   2 Limit speed of journal rotation and limit amount of rotation to several revolutions.
c. If a propulsion turbine must be rotated and lube oil flow is unavailable, take the following action:
   1 Add oil to all turbine and gear SFI feed lines (not necessary if lube oil system operated within 48 hours).
   2 Coat the turning gears with lube oil immediately before rotation.
   3 Rotate the propulsion train under no-load, no-torque conditions (not to exceed 1-1/4 turns of bull gear), followed by a 1-hour cooldown, followed by 1-1/4 turns of bull gear if necessary.

231-3.3 SHORTENED WARM-UPS FOR SUBMARINES

231-3.3.1 GENERAL. For submarine turbines a shorter warm-up is often desired to meet operational requirements. Note that the normal warm-up should be used whenever possible since it is less likely to cause rubs and problems and will result in longer service life for the turbine.

231-3.3.2 MAIN PROPULSION TURBINE. Main turbine warm-up options are described in the turbine technical manual or often in the Steam and Electric Plant Manual. Included are such warm-ups as normal, quick, rapid, reduced, after emergency propulsion motor (EPM), and emergency. If a main turbine has not been in standby and if a normal warm-up is impossible, the following shortened warm-up procedure may be used.

231-3.3.2.1 Rapid Warm-Up. Use this procedure if the turbines have been operating, have not stopped for more than 5 minutes, and have been on the jacking gear for less than 2 hours. The gland seal and exhaust system and the main condenser vacuum may or may not have been available during jacking.

a. Establish gland seal and exhaust and vacuum. With lube oil from cooler at least 80°F, disengage jacking gear and spin turbines once astern and once ahead, declutched if applicable and not to exceed the revolutions per minute (RPM) for this class of ship.

b. Place turbines in service, and answer bells cautiously and gradually.

c. If unusual vibration is noticed during warm-up, complete a normal warm-up in accordance with ship's operating procedures.

d. If unusual noise is heard during warm-up, stop turbines and investigate.

231-3.3.2.2 Reduced Warm-Up. Use this procedure if the main turbines have been operating, have not stopped for more than 5 minutes, and have been on jacking gear for less than 4 hours:

a. Establish gland seal and exhaust and vacuum. With lube oil from cooler at least 90°F, disengage the jacking gear and spin the turbines astern then ahead, declutched if applicable and not to exceed the RPM for this class of ship, for 8 to 10 minutes.

b. Place turbines in service and answer bells cautiously and gradually.

c. If unusual vibration is noticed during warm-up, complete a normal warm-up in accordance with ship's operating procedures.

d. If unusual noise is heard during warm-up, stop turbines and investigate.

231-3.3.2.3 Emergency Warm-Up. Use this procedure for tactical emergency only, not for training or for drills:

a. Place the propulsion turbines into operation without warming up. Build to ordered speed as gradually as conditions allow.

b. If unusual noise or vibration is noticed, respond as conditions allow.

231-3.3.3 SSTG SETS. SSTG set warm-up options are described in the SSTG technical manual. Such warm-ups include normal, reduced, and emergency warm-ups. Do not use emergency warm-ups for training or drills.

231-3.3.3.1 SSTG Reduced Warm-Up. Use this procedure if the turbines have been operating, the SSTG normal warm-up is impossible, and the SSTG has been jacked for less than 2 hours. Gland seal and exhaust or condenser vacuum may or may not have been available during jacking.

a. Establish gland seal and exhaust and vacuum.

NOTE

If the SSTG has been stopped for less than 5 minutes, proceed to step c.

b. Quickly crack open the throttle, and start the rotor spinning about 100 RPM. Trip and reset.

c. Open throttle slowly, and bring the RPM up slowly until governor takes control. Open throttle wide, and SSTG can take loading.

d. If an unusual noise is heard during warm-up, secure the SSTG and investigate.

e. If unusual vibrations are noticed during warm-up, reduce speed by 200 RPM and hold for 5 minutes before proceeding.

231-3.3.3.2 SSTG Emergency Warm-Up. Use this procedure for tactical emergency only, not for training or drills:

a. Place the SSTG into operation as rapidly as possible without warming up. Build to operating speed as gradually as conditions allow.

b. If an unusual noise is heard during warm-up, secure the SSTG and investigate.

c. If unusual vibrations are noticed during warm-up, reduce speed by 200 RPM and hold for 5 minutes before proceeding.

231-3.4 STANDBY

231-3.4.1 GENERAL. A turbine in standby should be ready to respond to all bells. The following defines a proper warm-up:

a. The lube oil is being circulated, and its temperature is from 90°F to 120°F.

b. The gland seal and exhaust system is operating.

c. The engines are being spun alternately ahead and astern at low speed every few minutes.

d. All water has been drained.

231-3.4.2 OTHER INSTALLATIONS. In some installations the manufacturer may recommend additional instructions or slight variations in the conditions listed in paragraph 231-3.4.1 These will appear in the applicable technical manual (in the operation section under standby condition) and should be followed.

231-3.5 MANEUVERING

231-3.5.1 HANDLING SPEED CHANGES. An increase or decrease in propulsion power is required to change ship speed, and this requires a change in main steam flow. Speed changes are initiated by orders from the bridge and carried out by the throttle operators moving the throttle in the direction, and to the opening, required to establish the new speed. The boiler and boiler support equipment either react automatically or are adjusted manually to follow and support the changes.

231-3.5.1.1 Setting of Speed Ordered. The approximate turbine first-stage pressure, HP or intermediate pressure (IP), required to produce a given number of propeller turns per unit time (ship speed in knots) can be established from design information. The actual pressure required will vary somewhat, depending on the specific installation, ship displacement, injection temperature, sea conditions, hull condition, and propeller condition. Average values can be determined from experience on individual propulsion plants and listed for use by the throttle operators, thereby simplifying the setting of speed ordered in operation.

231-3.5.1.2 Acceleration Tables. Standard rates of speed change are established by either the Type Command or the individual ship in the form of acceleration tables. These tables normally specify speed changes in knots/minute and (recognizing the disproportionate increase in power required for higher speeds) allow longer time periods at higher powers. Since power developed and first-stage pressure are proportional to steam flow, moving the throttle to produce a gradual and steady increase in first-stage pressure [(delta)psi/t a constant] will provide a smooth transition and avoid unnecessary acceleration loading on the equipment. Where the number of active turbine stages is varied as power is changed, discontinuities in some pressure-flow relations occur. These must be considered when creating a table of pressures and establishing procedure. This could mean a change in monitored pressure, from the HP first stage to the IP first stage for propulsion turbines of HP-IP series-parallel design. Pressure gages are provided at as many points as necessary to cover the speed range. Monitoring the highest pressure still responding proportionally to flow changes is always advantageous because this measurement is inherently most nearly accurate.

231-3.5.1.3 Rapid Throttle Operation. Under emergency conditions the throttles can be opened more rapidly. Throttle position can vary from that needed to produce a twice the standard rate of speed change to that requiring full capability of the equipment. Keep demand consistent with the boiler(s) steaming rate; otherwise, the main steam pressure could drop to a point where the boiler will not recover. Establish a limiting pressure below which steam pressure is not allowed to drop, and instruct the throttle operator to not open the throttle any further if the low point is reached.

231-3.5.1.4 Crash Astern. A crash astern (or back emergency maneuver) is a quick reversal from ahead power to astern power to stop the shaft or stop the ship. For a surface ship the astern valve should be opened as the ahead valve is being shut. This will prevent harmful boiler transients. For a submarine the ahead valve should be shut, then the astern valve opened. Turbines and line shaft equipment are designed to perform a crash astern.

231-3.5.2 ASTERN OPERATION. In all geared-turbine drives, the astern turbine shares a common turbine shaft with the ahead turbine and is located in the ahead turbine exhaust space. This placement of the astern turbine represents the best compromise between some disadvantages (in terms of operating flexibility) and advantages (in weight, space, and cost) over alternatives. The ramifications of this choice are discussed because, to some extent, maintaining design thermodynamic performance and turbine structural integrity depends on the judicious use of the astern turbine. Remarks apply mainly to turbines using highly superheated steam in cross-compound-type arrangements. Non-nuclear surface ships have cross-compound arrangements and use superheated steam. On all surface ships the LP turbine contains the astern section.

231-3.5.2.1 Astern Operation Impact on Temperature. In these designs, astern operation produces large temperature changes in the LP turbine that can cause overheating of either (or both) the HP and LP turbines. The largest changes in internal axial and radial clearances of the LP turbine are also associated with astern operation. Typical performance and characteristics are described in paragraphs 231-3.5.2.2 through 231-3.5.2.5.3.

231-3.5.2.2 Factors Affecting Heating Rate. The backward rotation of the ahead blading during astern operation churns and tends to pump the ambient steam, producing heat from friction in the ahead section. The amount of heat generated in each stage is a function of blade ring diameter, blade height, clearance, RPM, and steam density. Because of the numerous stages in the ahead elements of the LP and HP turbines, the amount of heat generated is considerable. Under steady-state conditions at constant astern speed, the temperature of steam exhausted from the astern turbine and the heat generated from windage will be approximately constant and the metal and steam temperatures of the turbines will rise until heat losses equal the heat input.

231-3.5.2.3 Time and Temperature Limitations. Test of a CG (formerly DLG) design turbine with high inlet steam temperature of 940°F showed that the turbines could not be run to this equilibrium condition on full-power astern. Instead, the material temperature limit was reached in approximately 1 to 2 hours, with the longer time depending on better-than-design vacuum in the condenser.

231-3.5.2.3.1 Note that the times referred to in paragraph 231-3.5.2.3 apply only to the particular turbine tested and are presented only as an example. Heating of the HP or LP turbine and the time taken to reach its design temperature limit during astern operation will depend to a great extent on what ahead power was being developed before going astern, since the heating effect from windage increases the temperature from the base condition established by the previous operation. Ahead nozzle valve leakage (if present during astern operation) aggravates the heating in the HP turbine because of work being done on the already hot entering steam.

231-3.5.2.3.2 Shaft speed (SRPM) astern limitations are normally given a continuous (or steady-state) SRPM allowable for any length of time, and also a transient SRPM (higher than the continuous SRPM) allowable only for a short length of time (typically 5 minutes) to support emergency astern operation to stop the ship. These are normally given in the turbine technical manual. It may not be possible to reach the astern transient SRPM when stopping the ship or when trying to operate astern at maximum astern SRPM. This can be due to hull or propeller characteristics or the steam pressure or condenser vacuum available. Not reaching this transient SRPM is not a problem and it does not have to be demonstrated; any SRPM over the continuous SRPM is acceptable.

231-3.5.2.3.3 Actions that will slow the heating of turbines during astern operation are:

a. Maintaining condenser vacuum at the best obtainable level
b. When permitted, lowering inlet steam temperature for separately fired superheater in the boiler
c. Reducing astern speed as low as practical
d. Reducing the duration of astern operation
e. Activating exhaust hood sprays

231-3.5.2.3.4 Refer to the technical manual for each particular propulsion set for more definitive information on actual temperature limits and identification of instruments that are to be temperature monitored.

231-3.5.2.4 Casing Distortion. Temperature differentials in the LP turbine normally cause the casing to hog, or hump, upward. This distortion tends to be at a maximum during prolonged astern operation. During astern operation, the relatively stagnant conditions in the ahead blading permit hot steam to collect at the top of the inner casing and create significant vertical temperature gradients along the longitudinal centerline. The tendency is further aggravated by:

a. The astern exhaust steam being swept (or pumped) by backward-turning ahead blades into the inner wheel case
b. The direct loss of heat by radiation from the lower inner wheel case to the condenser
c. Restraint of the cooler condenser on the hot turbine casing

231-3.5.2.4.1 The variables affecting hog of the LP turbine casing are astern steam inlet temperature, astern power, casing configuration (obstructions and clearances), and duration astern. At full-power astern for periods of approximately 1 hour, larger LP turbines have experienced hogs of approximately 0.050 inch. These motions are certainly significant when compared with the initially set shaft packing clearances of 0.010 to 0.020 inch so that rubbing of packings could be quite severe in the casing-distorted condition. Distortions are minimized by the same actions recommended for avoiding turbine overheating in paragraphs 231-3.5.2.3 through 231-3.5.2.3.3.

231-3.5.2.5 Differential Expansion. The low-pressure turbine blading clearances in any particular turbine are based on experience and on the computed difference in expansion and cooling rates of the rotor and casing expected under the most severe operation. These clearances should be small for least leakage and best efficiency, but they must also be large enough to avoid rubbing during transient conditions. The greatest changes in clearance are known to be associated with ahead-to-astern and astern-to-ahead maneuvers. This can be explained on the basis that, in most LP turbine designs, the casing heat transfer area-to-weight ratio is higher than the same ratio for the rotor; the casing, therefore, grows or shrinks more rapidly with temperature changes. The temperature level differs considerably in ahead and astern operating modes and an average temperature change of 300°F from one mode to the other is common. Typical effects of operation on clearances are described in paragraphs 231-3.5.2.5.1 through 231-3.5.2.5.3.

231-3.5.2.5.1 The starting condition is the turbine warmed-up condition. By admitting hot steam for astern operation, the casing initially heats more rapidly than the rotor. This is especially true for reaction-type turbines where the drum rotor is more massive than the casing. Because of the rotor length-to-radius ratio, the largest clearance changes occur in the axial direction. For designs with the thrust bearing forward, the casing moves aft with respect to the rotor. At various positions along the longitudinal axis, the amount of movement is approximately proportional to the distance from the thrust bearing and the overall differential growth. The clearance diagram for the turbine shows how the various blading clearances will be affected by such a casing motion. If the astern operation continues indefinitely, temperatures would eventually stabilize and casing and rotor growths would be about equal. In an extended astern period, differential growth would reach a maximum (with resultant minimum axial blading clearance) and then decrease with reduction or cessation of astern power, eventually reaching approximately zero at thermal equilibrium. In any period of ahead operation following astern operation, the casing would cool more quickly and shorten, relative to the rotor, and again produce minimum blading clearance, but in the short-casing, long-rotor condition.

231-3.5.2.5.2 Many factors determine whether clearances will close to the point of rubbing during ahead and astern maneuvers. The rubbing would most likely occur when extended astern operation at high-steam temperature and power is followed by ahead operation at other than the most efficient ahead turbine condition. Conversely, the rate of differential expansion in the ahead operation following an astern period can be slowed by operating at low ahead speeds or best efficiency (preferably below 15 knots) or by stopping the shaft (which will produce a bowed rotor). The trend can be reversed by again reversing the turbine.

231-3.5.2.5.3 The difficulties described can usually be avoided by reducing superheat (only in boilers with separately fired superheaters) during the astern period. Some newer ships with no independent control of superheat have LP turbines equipped with differential expansion indicators. These should be monitored during maneuvers and action should be taken to avoid exceeding the differential expansion limits specified in the applicable technical manual.

231-3.6 LOCKING AND UNLOCKING MAIN SHAFT

231-3.6.1 GENERAL. Propeller shafts in multishaft turbine-driven ships can be locked to prevent shaft rotation. Where way must be kept on the ship but the rotation of the propeller shaft may cause or compound damage to the propulsion machinery, the shaft should be locked as quickly as possible. For detailed instructions when locking and unlocking the shaft in submarine applications, see NSTM Chapter 241, Propulsion Reduction Gears, Couplings, and Associated Components, and the appropriate reduction gear technical manual.

231-3.6.2 PERSONNEL QUALIFICATIONS. Each engine room watch and each general quarters watch team should be qualified in underway locking and unlocking procedures (NSTM Chapter 079, Volume 3, Damage

Control - Engineering Casualty Control). Because of unusual turbine heating associated with the locking procedures, perform only those exercises required to maintain qualification, particularly in plants operating with highly superheated steam and no superheat control, where nonrotation of the shaft in a hot plant for periods exceeding 3 to 5 minutes could bow the rotor.

231-3.7 DISCONNECTING TURBINE ON SINGLE-SCREW SHIPS

231-3.7.1 GENERAL. An emergency means of ship propulsion, in single-screw, geared-turbine-driven ships, is provided when one of the two turbines installed is disconnected and propulsive power is supplied by the other. The necessity for emergency ship propulsion assumes a casualty that will not allow the rotation of one turbine. In installations where the two turbines are identical, with each turbine receiving steam from the main steam line and exhausting to its own condenser, the procedure for disconnecting a turbine is relatively simple and can be quickly accomplished. With cross-compound turbine sets, however, it is necessary to admit main steam directly to the LP turbine when the HP turbine is disconnected or to divert steam from the HP turbine directly to the condenser when the LP turbine is disconnected. These arrangements involve considerable work and time-consuming operations.

231-3.7.2 PROCEDURE. Study and follow the detailed instructions contained in the manufacturer's technical manual and in the shipbuilder's instructions when disconnecting a turbine. The special equipment required for the operation is normally listed in the equipment technical manual. Physical location of parts should be known and verified periodically so time required to accomplish the singling-up operation, if it becomes necessary, will be minimized. The general procedure for disconnecting a turbine is to:

1. Stop the main shaft at a rate indicated by conditions of the situation. Use the turbine for reversing torque and braking action, if necessary.
2. Engage jacking gear and apply shaft brake. Secure all steam to the turbine. Do not turn over with turning gear, but maintain lube oil flow, if permissible, until temperature rise across hottest bearing is 15°F or less.
3. Wire, chain, tag, or otherwise make arrangements that will prevent accidental opening of throttle or guardian valves when disconnecting the turbine.
4. Disconnect inoperative turbine by disengaging or removing high-speed coupling distance piece, as appropriate.
5. Lock inoperative turbine against rotation.
6. Blank gland seal, lube oil, or main steam lines to locked turbine, if and as required.
7. Resume operation with single turbine, taking necessary precautions associated with bowed rotor, as discussed in paragraphs 231-3.11 through 231-3.11.2.6.
8. Keep operating variables and loads within limits specified in the technical manual. Where specific limits are not given, limit speed of propeller to 60 percent of normal full-power RPM.

231-3.7.3 PRECAUTIONS. Special precautions are required in singled-up operation because:

a. Accidentally admitting main steam to a disconnected turbine could cause overspeeding and turbine destruction.
b. Improperly adjusting desuperheating water, when required in single-LP-operation, could result in either overheating LP ahead element from high temperature or mechanical damage from water slugging.

c. Improperly depressurizing HP turbine exhaust, when LP turbine is bypassed, could overload the condenser and low vacuum.

d. Higher torques developed by a turbine at lower speeds make it possible to overload the turbine in use (or that half of the reduction gear that is transmitting power) within steam flow-passing capability of the turbine.

e. Disconnecting LP turbine prohibits use of the astern turbine.

f. Improperly locking disconnected turbine rotor could cause unexpected rotation due to inertial motions from sea action (or leakage of steam into turbine where steam and gland seal lines are not blank-flanged).

g. Improperly blanking gland seal and lubricating oil connections on the disconnected turbine could cause unexpected, dangerous leakages of gland sealing steam into the locked turbine.

231-3.7.4 PRECAUTIONS WHEN RECONNECTING. When a plant has been singled up and run for any length of time in this condition, the operating element has probably experienced higher-than-normal temperatures, which could cause flange bolts to relax, with resultant steam leakage. This leakage may not be apparent under the singled-up condition, but may first appear when two-turbine operation is resumed. Should steam leakage occur, slugging-up on casing flange bolts (which should not exceed the elastic limit of the bolts) may temporarily stop the leak. Similarly, pumping the gunning groove, if provided, may stop the leak. If these precautions fail, remaking the flange or renewing the bolting will probably be necessary.

231-3.8 WINDMILLING

231-3.8.1 EMERGENCY CONDITIONS. Under emergency operating conditions one turbine in a twin single-casing arrangement that has the rotor shafts coupled to the high-speed pinions of a double-reduction gear arrangement can be allowed to windmill. Windmilling is achieved by securing the main steam to the turbine. This is an alternative to single-turbine operation where the turbine is disconnected from the reduction gear (paragraphs 231-3.7.1 through 231-3.7.4 and 231-5.5 through 231-5.5.2). The abnormal turbine heating produced during windmilling is discussed in paragraphs 231-3.5.2.2 and 231-5.3.2.

231-3.8.2 PRECAUTIONS. General precautions to be taken when windmilling are:

a. Do not exceed windmilling operating parameters as specified in the turbine equipment manual.

b. Provide gland sealing steam to the windmilling turbine.

c. Maintain best practical main condenser vacuum.

d. Maintain proper lubrication.

e. Activate turbine exhaust boot sprays.

231-3.9 SECURING

231-3.9.1 The procedures for securing a main propulsion turbine and an SSTG turbine are similar. To secure the SSTG turbine, however, shift the load to another machine by reducing the load on the generator and trip the generator circuit breakers in accordance with NSTM Chapter 310, Electric Power Generators and Conversion Equipment. Then securing either the main propulsion or the SSTG turbine normally includes closing the throttle that admits main steam to the turbine; opening drains; and shutting down oil, gland, and vacuum systems. The order in which these are to be accomplished and the amount of time various systems should remain operating during cooldown are given in the equipment manual and the Engineer Operating Sequencing System (EOSS). Follow established procedures closely to retain the short-term ability to restart the turbine quickly and to resist turbine corrosion during idle periods over the long term.

231-3.10 TURBINE PERFORMANCE

231-3.10.1 EFFECTS OF PACKING CLEARANCE.

231-3.10.1.1 Gland Packing. Packing is used throughout the turbine to control either steam or air leakage. Excessive gland packing wear will cause steam to leak into the engine room or loss of condenser vacuum because air enters the main condenser. Packing can be examined by lifting the small end covers without lifting the main turbine covers. Generally, steam would leak into the engine room from the HP turbine and vacuum loss would be associated with low-pressure turbine glands. Understanding pressure variations and leak behavior during operation through the power range will simplify finding the leak. The gland seal and exhauster system should generally be the first suspects and should always be thoroughly checked whenever a leak or vacuum loss occurs.

231-3.10.1.2 Interstage Packing. Interstage packing controls and limits leakage of steam that does not pass through blading and deliver work. The differences in performance at light load and high power are believed to be related to the disturbing effect of the re-entering interstage packing leakage flow on the low stage throughflow. The performance change at low power is too great to be attributed only to an increase in leakage area. The majority of change is in the HP turbine in a cross-compound turbine set. This is expected since the design leakage clearance represents several percent of the active steam path flow area at the high-pressure end of the HP turbine and less than 0.1 percent at the LP turbine exhaust end.

231-3.10.1.3 Clearance Philosophy. Two different philosophies exist concerning the establishment of initial design clearances. One philosophy is to set clearances closer than experience has shown can be reasonably maintained and then allow them to rub and wear the amount necessary to suit transient thermal motions of the casing and rotor. The other philosophy is to set initial clearances wide enough so that no rubbing will occur. Although it may seem that the former philosophy will automatically result in the lesser leakage, the rubs can be so severe as to cause the rotor to overheat locally, and bow and vibrate so that the packings wear excessively. Also, the seal teeth tend to mushroom when rubbed. This increases both the flow coefficient and the leakage flow. The designer therefore attempts optimization by allowing some small amount of rubbing.

231-3.10.1.4 Conclusions Based on Experience. Experience in testing LP turbines has shown that the casing tends to hog during astern operation, caused by restraint of the hot turbine by the cooler main condenser. LP turbine horizontal joint flanges have been shown to bow upwards as much as 0.060 inch. With diaphragms supported at the flange, the diaphragm packings are carried up into the rotor with hog, and are worn an amount equal to the casing hog less initial clearance. The manufacturer sets diaphragms low to partially offset the effect of casing hog. Some distortions have been measured in HP-IP and bypass flow HP turbines, and although considerably less than that on LP turbines, average packing wear in the order of 0.005 to 0.010 inch has been measured. The discussion in paragraphs 231-3.10.1 through 231-3.10.1.4 is summarized here:

a. Some packing wear is normal in a turbine. Hog from extended astern operation can cause LP turbine interstage packing wear as great as 0.050 inch. HP turbine wear of 0.005 to 0.010 inch from normal ahead operation is not uncommon.

b. Relatively large amounts of wear in LP turbine packing has less effect on performance than wear occurring normally in HP turbines.

c. Falloff in performance due to normal packing wear is less than 1 percent at full power.

d. Packing wear is a result of thermal distortion and can be minimized by correct preheating of the turbine.

e. Mushroomed packing teeth pass more leakage flow. Reused packing should therefore have teeth restored to original thickness by hand-scraping or other suitable means.

f. Setting packing clearance too close is self-defeating because a heavy rub will cause packing and shaft to grow into each other from heat developed. This will cause greater wear than wider initial clearance and light rub.

g. Setting packing clearance too close can cause vibration from local heating and bending of the turbine rotor.

h. Steam leaks and vacuum problems in most cases are a result of incorrect gland seal and gland exhauster system performance, rather than worn packing.

i. An increase in gland packing and interstage packing radial clearances of up to 0.005 inch greater than design clearance due to undersized packing journals does not require special packing rings.

231-3.10.2 CONTROL VALVES. The number and type of steam-admission valves depend on the type of turbine and its intended purpose, as discussed in paragraphs 231-2.3 through 231-2.3.12.3. The control valves are sequentially operated by manual or power means to regulate steam flow and thereby power output to the gear. Valve openings and closings are timed to ensure maximum turbine efficiency. These settings constitute valve points, which are discussed in paragraphs 231-3.10.2.1 through 231-3.10.2.3. Set throttle valves in accordance with the original equipment manufacturer (OEM) control drawing.

231-3.10.2.1 Riding Valve Points. A valve point is defined as the particular position of the turbine valve gear where one valve is at the extreme open position, just short of the cracking (opening) of the next valve to open in sequence. This position corresponds to one of a series of optimum performance points of the turbine. The operator shall be familiar with the bar position or shaft RPM that produces the optimum turbine performance points. Every precaution shall be made to ensure that a valve is not being held in a cracking position. If a valve is held in a cracking position, severe throttling losses will occur across the valve, resulting in a loss of turbine performance. Also, the high-velocity steam produced by the throttling will cause the valve seat and disk to erode or wear more rapidly. The intent is to save the valve seats by not having a valve remain in a cracking position.

231-3.10.2.1.1 The cracking point of the first valve to open can be located within a reasonable range by using a listening rod or other device at the steam chest to detect the onset of hissing steam. Adjust the port and starboard ahead turbine throttles to crack at the same (synchronized) point. Also synchronize the port and starboard astern throttles. This allows the turbines to track or share load at low power levels and prevents a misadjusted throttle from allowing steam into the turbine when secured.

231-3.10.2.2 Throttle Tracking. In a submarine propulsion plant where two identical steam turbines operate at the same steam conditions to produce half the total power, the first-stage pressure is monitored to verify that each turbine is operating within design and equally sharing the load. Maintaining the design first-stage pressure on each turbine as power levels change is called throttle tracking or balancing. Balancing cannot be done dockside on ships without a main propulsion clutch.

231-3.10.2.3 First-Stage Shell Pressure Differences. Do not operate any turbine outside its first stage shell pressure curve, which is given in the turbine technical manual. In submarine propulsion turbines the throttle tracking or the first-stage shell pressure difference (port and starboard) should not exceed 15 psid (or 5 percent of full-range pressure for Delaval) at any power level. This ensures that maximum power from both turbines is being used and that the reduction gear is not being unevenly loaded. If resetting is necessary, disconnect the throttle control linkage splined coupling and lower the first-stage shell pressure of the leading turbine. In an availability, do not disconnect splines, but follow OEM instructions for valve setup (for example, the GE N-link adjustment). Verify that both throttles will shut completely, that excessive closing force is not being used, and that cracking points are still synchronized. If cracking points cannot be synchronized after throttles have been set to track to the valves described above, then the valve lifts are improperly set. The turbines may still be run, but the cracking points must be set and temporarily readjusted when the shell pressure spread exceeds that of the valves described above. The valves should be worked at the next availability.

231-3.10.2.4 Astern Bowl Pressure Differences. In submarine propulsion turbines the astern bowl pressure gage difference (port and starboard) should not exceed 25 psid (or 5 percent of full-range pressure for Delaval) at any power level. Do not operate any turbine past its astern bowl pressure limit, which is given in the turbine technical manual.

231-3.10.2.5 Comparing Superheat with Saturated Steam. Superheat control may or may not be available in a particular propulsion plant, depending on the type of main boiler installed. With separately fired superheaters, the main steam temperature can be varied from saturation to the design upper limit. Except for special requirements of astern operation, it is desirable to operate at, or as close as possible to, design temperature. Special requirements of astern operation are discussed in paragraphs 231-3.5.2.3 through 231-3.5.2.3.3. Harmful effects of operating at lower superheat than specified are as follows:

a. Power capability is reduced by approximately the same percentage as the reduction in absolute temperature.

b. Turbine efficiency is reduced by high moisture loss in stages operating in the wet-steam region. (Available energy is lost as steam condenses in the stage.)

c. The tendency for erosion damage to rotor blades operating in the wet-steam region is increased.

d. The possibility of forming condensate in the main-steam piping is increased.

231-3.10.2.5.1 Special drainage provisions and special corrosion resistant materials are used in turbines that normally operate with saturated steam at the inlet. Submarines and nuclear surface ships operate with saturated steam.

231-3.10.3 EVALUATING MECHANICAL PERFORMANCE OF TURBINES. The propulsion and SSTG turbine mechanical problems commonly experienced can usually be avoided by:

a. Keeping contaminants in steam and lubricating oil to a minimum

b. Properly heating and cooling the turbine in operation

c. Properly drying turbines before shutdown

231-3.10.3.1 Maintenance for Longevity. By following the above the steam turbine will operate for 30 years. Mechanical difficulties are normally indicated by rubbing sounds, high-bearing temperatures, or abnormal vibrations. Actions to be taken when abnormalities occur are discussed in paragraphs 231-3.10.3.2.2 through 231-3.10.3.5. A vibration troubleshooting and information guide can be found in Table 231-3-1. Vibration levels are discussed solely in terms of mechanical reliability; special low-level noise requirements are covered in other documents.

231-3.10.3.2 Shaft Vibration, Measurements, and Significance. Propulsion and SSTG turbine rotors are precisely manufactured components. At the time of manufacture, the rotors are balanced to the degree that rotor vibration in the assembled machine is a fraction of a mil (0.001 inch) in amplitude. Specification limits that control the end product vary, depending on whether the application is considered noise critical. In all cases, however, the forces represented by the residual imbalance and speed are small. A moderate increase in the vibration level is expected in the ship installation from additional imbalances introduced by the driven unit and from some small rotor distortions at operating temperature. In the ship, however, these levels are expected to remain well below 1 mil (paragraph 231-8.13).

231-3.10.3.2.1 Vibration in a turbine is a positive indication that the unit is not in proper working condition. As soon as this condition is noted a thorough investigation should be made to determine the cause of the trouble. If the trouble is not remedied immediately and defects are allowed to accumulate, bearing and packing clearances become excessive, with consequent loss of oil and steam. The bearings and packing are soon ruined, and, if the turbine is kept in operation, further trouble may develop, completely disabling the unit.

Table 231-3-1 VIBRATION IDENTIFICATION GUIDE

Cause	Frequency Relative to Machine RPM	Phase-Strobe Picture	Amplitude	Notes
Imbalance	1 x RPM	Single steady reference mark	Radial-steady proportional to imbalance	Common cause of vibration.
Defective antifriction bearing	10 to 100 x RPM	Unstable	Measure velocity 0.2 to 1.0 in/sec radial	Velocity largest at defective bearing. As failure approaches, velocity signal will increase, frequency will decrease.
Sleeve bearing	1 X RPM	Single reference mark	Not large	Shaft and bearing amplitude about the same.
Misalinement coupling or bearing	2 X RPM sometimes 1 or 3 X RPM	Usually two steady reference marks - sometimes 1 or 3	High axial	Axial vibration can be twice radial. Use dial indicator as check.
Bent shaft	1 or 2 x RPM	1 or 2	High axial	---------------
Defective gears	High RPM x gear teeth	-----------	Radial	Use velocity measurement.
Mechanical looseness	1 or 2 x RPM	1 or 2	Proportional to looseness	Radial vibration is largest in direction of looseness.
Electrical	Power line frequency x 1 or 2 (3600 or 7200 RPM)	1 or 2 rotating marks	Usually low	Vibration stops instantly when power is turned off.
Oil whip	Less than RPM	Unstable	Radial-unsteady	Frequency may be as low as 1/2 RPM
Aerodynamic	1 x RPM or number of blades on fan x RPM	------------	------------	May cause trouble in case of resonance.
Beat frequency	1 x RPM	Rotates at beat rate	Variable at beat rate	Caused by two machines running at close RPM.

Table 231-3-1 VIBRATION IDENTIFICATION GUIDE - Continued

Cause	Frequency Relative to Machine RPM	Phase-Strobe Picture	Amplitude	Notes
Resonance	Specific criticals	Single reference mark	High	Phase will change with speed. Amplitude will decrease above and below resonant speed. Resonance can be removed from operative range by stiffening.

231-3.10.3.2.2 Experience has shown that a sudden increase in vibration level of a turbine that has been operating normally can be traced to one or more of the following causes:

a. Water being carried over from steam generators

b. Thrust or journal-bearing problems

c. Breaking of turbine blade(s)

d. Heavy rubbing of blading, carbon packing, labyrinth packing, or oil-seal rings

e. Driven unit (generator, gear) out of balance

f. Drive unit out of alinement

g. Turbine rotor out of balance

h. Loose or broken foundation bolts

i. Bent shaft

231-3.10.3.2.3 Depending on the cause, the vibration increase may be serious (and require opening the turbine for repair) or temporary (in that the cause can be eliminated and normal operations restored). This will usually be difficult to assess, and action taken should depend on the severity of the vibration. If vibration is heavy, the turbine should be shut down immediately and the jacking gear engaged. If the jacking gear will not turn the rotor or if a rubbing noise is heard while jacking, check the vertical positions of the journals at the bearings, the rotor axial position, and the thrust-bearing clearance. Also examine turbine internals (through access openings) for evidence of rubbing, broken parts, or any loose or foreign material.

231-3.10.3.2.4 If a question remains as to whether or not levels are acceptable, measure the vibration amplitude at the bearing caps with an instrument suitable for detecting levels in the range of 0.5 to 10 mils and capable of identifying the predominant frequency. Further action should depend on whether the vibration is related to the turbine (submultiple or multiple of turbine rotational frequency) or a frequency related to some other system component.

231-3.10.3.2.5 Moderate vibration increases can be assumed to be the result of a rub from a slightly bowed rotor. Speed should be reduced so as to reduce vibrations and held for 5 minutes. Re-establishing normal vibration will indicate a straight rotor, and the turbine should again be ready for unrestricted use.

231-3.10.3.2.6 A characteristic of most surface ships with geared main propulsion systems is that the main turbines will respond (vibrate) to propeller excitation at propeller blade-rate (propeller RPM x number of blades) frequency. The response characteristic is a function of the particular foundation stiffness and mass, and the resultant amplitudes will depend on the excitation force. This alternating force is in turn a function of load and propeller condition. The vibration is noticeable at high powers, and turbine response is principally in the axial direction.

231-3.10.3.2.7 Propeller blade-rate vibration amplitudes ranging from 0.005 to 0.010 inch are common and may be as high as 0.030 inch. Although operation at the highest levels have not proved harmful to the turbines involved, rapid high-speed flexible coupling wear is possible and operation at high amplitudes should be avoided.

231-3.10.3.2.8 The propeller blade-rate is a variable-frequency source of excitation with speed change. It can induce vibration in various items throughout the ship. Grating support structures and some piping sections have low-frequency response modes that will be excited by the propeller at high motions. These effects should not be mistaken for a turbine problem.

231-3.10.3.3 Bearing Temperatures and Limits. Temperature is the sole criterion available to the operator for judging the condition of individual journal bearings during turbine operation. Watchstanders, therefore, should monitor and log temperatures periodically so that any tendency toward bearing overheating can be detected early and corrective action taken. Bearing temperatures increase with speed and a change in speed.

231-3.10.3.3.1 The bearing operates over a normal temperature range. Proper operation of the turbine journal bearings can be ensured when temperatures at a given speed repeat to within a few degrees of those obtained when bearings were known to be in good condition. Temperature versus speed at standard speeds should therefore be established for reference. Note that the temperature of oil supplied to the bearing normally ranges from 120°F to 130°F. Comparisons should be made on the basis of equal inlet temperatures.

231-3.10.3.3.2 The temperature of oil out of the lube oil cooler and into the turbine may be less than 120°F or more than 130°F, according to conditions. Most propulsion and SSTG turbines may be started with 90°F oil and run with oil less than 120°F during normal startup or when the ship is in very cold water. Turbines may be run with oil inlet temperatures above 130°F; for instance, when coolers are partially fouled or when the ship is operating in water above the design seawater inlet temperature (typically 85°F) for the lube oil cooler. In these cases:

a. Bearing oil thermometer temperatures must be below 180°F.

b. Resistance temperature detector (RTD) babbitt temperatures must be normal if the bearings have resistance temperature elements (RTE).

c. Turbine controls should be tested and verified operational if they receive oil above 130°F from the lube oil cooler. These tests should be done prior to operating in water above the design seawater inlet temperature and should include turbine control valve operation, overspeed trip and standby pump autostart features.

NOTE

Whenever cooler outlet oil temperature cannot be maintained at, or below, 130°F (when the ship is operating in water at, or below, the design seawater inlet temperature), then the condition should be investigated and corrected if possible at the ship's next availability.

231-3.10.3.3.3 Thrust bearing temperatures can be expected to vary more than a few degrees at given speeds if a variable thrust force is imposed (such as that introduced by a high-speed flexible coupling in a geared propulsion system). The rotor-position indicator on most turbines should also be monitored periodically. The indicator and the thrust bearing temperatures can be used to assess the thrust-bearing condition. Rotor-position indicators that contact the moving rotor are susceptible to rapid wear if used improperly. They should be checked for accuracy when opportunity permits.

231-3.10.3.3.4 Bearing temperatures can be measured by two different methods with differing temperature limits. One method is to measure the bearing drain oil temperature by thermometer; the other is to measure the babbitt temperature by RTE. The drain oil-temperature limit is either a maximum temperature rise (inlet to outlet) of 50°F or a maximum temperature on the outlet thermometer of 180°F. Limiting temperatures for babbitt-embedded RTE's are 250°F for journal bearings and 270°F for thrust bearings.

231-3.10.3.3.5 While a 55°F temperature rise or a maximum thermometer temperature of 185°F is temporarily acceptable for continued operation of a ship at sea, it is abnormal and should be investigated during the ship's next availability. Where embedded RTE's are installed, their temperatures should be used to assist in evaluating the thermometer temperature.

231-3.10.3.4 Resistance Temperature Element. The RTE system provides central electrical readout of machinery temperatures and permits the use of an alarm that is normally set to sound when temperature reaches 20°F higher than that obtained during initial ship trials.

231-3.10.3.4.1 Practices that should be followed in setting temperature monitoring system alarms for propulsion and SSTG steam turbines are listed in Table 231-3-2. Deviations from these maximum settings must be approved by NAVSEA on an individual ship basis.

231-3.10.3.4.2 The initial settings shall be made before the first turbine operation. The final settings shall be made before the Inspection Survey (INSURV) trial or final postoverhaul trial if an INSURV trial is not run. The final settings shall be based on the highest values observed from the ship sea trials or shipbuilder test data taken during all previous trials.

231-3.10.3.4.3 For some propulsion turbines, the highest bearing temperature may occur at less than flank speed or during a turn because of bearing and turbine configurations. Run tests at increasing speed increments and allowable rudder angles. For SSTG sets RTE settings may be made according to Table 231-3-2 during dockside testing at 0 to 100 percent load or during sea trials, whichever gives the highest temperature reading.

231-3.10.3.4.4 The setpoints provided in Table 231-3-2 are to be used for propulsion and SSTG steam turbines if specific setpoint instructions are not contained in the Steam and Electric Plant Manual, if applicable, or in the component technical manual. The alarm settings shown in Table 231-3-2 are to be made with approximately 130°F oil discharging from the lube oil cooler and normal clearances in installed bearings. If a bearing or rotating element is replaced, check the bearing temperature alarm settings on the affected bearing(s) and reset in accordance with Table 231-3-2.

231-3.10.3.4.5 Ships equipped with RTE systems have a drain-oil thermometer in each bearing to permit an independent check of bearing condition. Note that the two measurements will not agree and each should be compared with either its applicable limit in Table 231-3-2 or, preferably, previous temperatures noted at the particular speed.

Table 231-3-2 MAXIMUM ALARM TEMPERATURE SETTINGS

Thermocouple and RTE Sensing Location	Monitor Alarm Setting	
	Initial (°F)	**Final (°F)**
Bearing babbitt	Journal bearing 250	Journal bearing. Set each bearing alarm 20°F above the bearing maximum running temperature observed during trials, or at 250°F, whichever is lower.
	Thrust bearing 270	Thrust bearing. Set each shoe's alarm 20°F above that shoe's maximum running temperature observed during trial or at 270°F, whichever is lower. (One shoe each side of bearing.)
Bearing oil drain line	180	180
Sight flow fitting	180	180

231-3.10.3.4.6 The RTE system responds more quickly to temperature changes and will give earlier warning of any bearing overheating. Any bearing temperature, measured by RTE or thermometer, that continues to rise after ship speed and oil inlet temperature have stabilized should be considered abnormal (despite being within specified limits) and should be investigated. Reducing turbine speed quickly reduces heat input into the bearing and that will most likely prevent a bearing wipe or limit bearing damage.

231-3.10.3.4.7 RTE alarm settings are made during the ship's construction or overhaul trials. Spurious RTE alarms after delivery can be caused by the alarms being set too low for all possible operating conditions or by an actual problem in the bearing such as bearing wipe, bearing misalinement, or babbitt irregularities near the RTE. Note that RTE's do not typically fail by drifting in temperature, but by electrical opens or shorts. Note also that splices are not recommended in RTE leads since they may cause changes in resistance and therefore changes in temperature indication. When a main turbine bearing RTE alarms, slow the ship as necessary and note the RTE temperature. Verify no damage by observing that temperature has stabilized, bearing oil flow is satisfactory, strainer has not collected babbitt, and turbine operates without unusual noise or vibration. When lack of damage is verified, recreate the alarm conditions and note the alarm temperature and stabilization temperature. A bearing much hotter than corresponding bearings may have alinement or babbitt problems. If no problems are seen, the RTE alarm may be reset per Table 231-3-2. If no problems are seen but the RTE has obviously failed, continue to operate the turbine and monitor bubbler thermometer temperatures at normal intervals; then disassemble to investigate at a convenient shore availability. If the RTE is faulty, replace the bearing or bearing pad. Bearing disassembly, if necessary, is not recommended at sea, but on return to port. Technical assistance (see paragraph 231-6.1.3) is recommended. For RTE's in SSTG's follow the same limits and procedures, but do not lower turbine speed without properly transferring electrical load.

231-3.10.3.5 RTE Operational Alarm Setting. This section applies directly to submarine steam turbines as described in 231-1.1.1 and it may be used for guidance in setting RTE alarms in surface ship steam turbines and all steam turbine driven gears and line shaft machinery. During the operating cycle, if it becomes necessary to redetermine and reset journal or thrust bearing RTE alarm set point(s), due to bearing replacement, repair or an RTE alarm verified to be previously set too low, the following procedures are to be used:

CAUTION

If abnormal bearing wear or alinement is suspected or bearing oil flow is abnormal, the conditions will be thoroughly investigated and resolved or corrected BEFORE alarm set points are adjusted. If in doubt, request technical assistance.

CAUTION

If, during these procedures, rapid RTE temperature increase occurs or the maximum RTE set point is exceeded, respond to a hot bearing casualty then perform troubleshooting and corrective maintenance in accordance with the guidance of paragraph 231-3.10.3.4.7.

231-3.10.3.5.1 A bearing RTE alarm set point should be reset to a value 20°F above the new baseline temperature but not to exceed the values specified below. Lube oil cooler outlet temperature should be maintained at 130°F. Bearing RTE alarm set points should be set so that:

a. Journal bearing alarms are not set above 250°F.

b. Thrust bearing (excluding line shaft thrust bearing) alarms are not set above 270°F.

c. Line shaft thrust bearing alarms are not set above 250°F.

d. The new baseline temperature for propulsion turbine journal bearings will be the highest temperature determined by answering ahead full bell, then incrementally increasing shaft RPM by 5 RPM increments while maneuvering using maximum safe rudder angles to both port and starboard. Record the highest temperature seen at each increment until the ship has achieved a flank bell.

e. The new baseline temperature for SSTG journal bearings will be the highest temperature seen during a one hour run with the SSTG nearly fully loaded. The new baseline temperature for SSTG thrust bearings will be the highest temperature seen with the SSTG unloaded, nearly fully loaded and with maximum safe rise and dive angles to add the effect of gravitational forces.

f. The new baseline temperature for other journal bearings and ahead thrust bearings will be the highest temperature seen on a one hour minimum, submerged ahead flank run.

g. The new baseline temperature for propulsion turbine, line shaft and reduction gear astern thrust shoe bearings will be the highest temperature seen during a surfaced back emergency bell for five minutes in calm water.

231-3.10.3.5.2 Data from the above tests should be recorded in the ship's operating logs, addendum, or bearing logs, with the new alarm set point(s) noted and retained for future reference.

231-3.10.3.6 Blade Vibration. Most blade failures result from resonant blade vibration. The vibration occurs in a rather narrow speed range that cannot be precisely defined. Such vibration is normally impossible to detect by external measurements, and there is no indication of trouble until blade or shroud material actually comes adrift. Such material does not come through the casing or travel downstream in pieces large enough to cause catastrophic damage. Depending on the turbine design, the broken blade may fly outward and become lodged between stationary parts, may fall to the lower part of the case, or may rapidly disintegrate by rubbing the casing at high speed or by being pounded to bits between rotating and stationary blading. Generally, only small shot-like pieces pass into the next stage. Damage, in the form of extensive nicks, is usually limited to the failure stage and the next stage downstream. The resulting noise indicates trouble, but since the process of breaking up debris is rapid, the noise will be of short duration. If an unusual noise is heard, inspect the turbine through existing accesses at the earliest opportunity for any sign of blade damage or debris. If an LP turbine casualty is suspected, also check the condenser hotwell for debris. A failure may go undetected if no noise is heard or if the resulting rotor imbalance is not great enough to increase shaft vibration noticeably. The fact that a given design has operated for many years without a blade failure does not eliminate the possibility of a future failure. Erosion, corrosion, or damage caused by the passage of foreign material may, in some cases, provide a place for the start of a fatigue crack that would not otherwise develop. Fatigue cracks can initiate in the root, base of the blade, midblade, or tenon.

231-3.11 STRAIGHTENING A BOWED ROTOR

231-3.11.1 REASONS FOR BOWING. Several conditions can produce the transient-type vibration associated with bowed-rotor operation. The direct cause of the bowing is a vertical temperature difference that produces distortion from unequal expansion, similar to that discussed in paragraphs 231-3.2.2.4 through 231-3.2.2.4.6. The conditions that are likely to produce bowing, the mechanisms involved, the characteristics, and actions to be taken are listed in Table 231-3-3.

231-3.11.2 PROCEDURE FOR STRAIGHTENING ROTOR. The intent of the procedure for straightening a bowed rotor is to eliminate the temperature gradient. To equalize temperature, the procedure will result in a transfer of heat to the cool side (or taking heat away from the hot side) of the rotor. The transfer can be accelerated by keeping the temperature as high as permissible in the turbine and using the highest steam velocity and density for the best heat transfer coefficient. Vacuum and speed should be reduced to bring vibration to moderate levels. As a rotor straightens, the remaining abnormal vibration will gradually disappear and normal vacuum and speeds can be resumed.

231-3.11.2.1 Rotor-straightening procedures are discussed in paragraphs 231-3.11.2.2 through 231-3.11.2.6 in the order of least to most chance of incurring additional damage, but in reverse order of minimum time for straightening. The procedure in paragraph 231-3.11.2.4 probably offers the best compromise.

231-3.11.2.2 If the shaft can be locked or the ship stopped for an extended period, securing all steam to the unit and allowing the rotor to cool naturally to ambient temperature will straighten the rotor with least risk (rubbing) to the turbines.

231-3.11.2.3 On multishaft ships, a safe method of removing a bow and straightening a rotor is to trail, or windmill, the bowed shaft, while underway on other shaft(s) at a speed that does not exceed acceptable vibration in the bowed unit. This procedure provides rotation in the presence of heat produced by frictional windage in the windmilling unit at low vacuum or atmospheric pressure.

231-3.11.2.4 If it is necessary to keep way on the ship, the shaft may be kept operating but the speed must be reduced to the point where vibration is largely eliminated. The vacuum produced by the second-stage air ejector alone should also be reduced, and speed should be held until residual vibration disappears or the vibration amplitude returns to the normal level for that speed. The rotor will then be straight, and normal speeds can be resumed.

Table 231-3-3 TEMPERATURE GRADIENTS IN BOWED ROTORS

Condition	Mechanism of Producing Temperature Gradient	Characteristic	Action
Rotor idle in hot casing	Natural top to bottom ambient casing temperature gradient in idle steam atmosphere causes greater heat transfer and higher temperatures at the top of the rotor.	Degree of vibration when rotated proportional to amount of time rotor was idle; vibration level decreases gradually at constant speed.	See paragraphs 231-3.11.2 through 231-3.11.2.6.
Water slug entering turbine	Rotor is chilled locally by water, producing severe gradient; distortion and vibration from hard mechanical contact between rotor and fixed casing parts.	Onset of vibration is almost instantaneous and severe; may be accompanied by steam leakage at casing joints and main steam line flanges.	Immediately shut down turbine. Use astern turbine if necessary. Allow turbine to cool down. Check for freedom of jacking. Straighten as discussed in paragraphs 231-3.11.2 through 231-3.11.2.6.
Heavy packing rub	High point on rotor heats locally from contacting shaft packing. Tends to be self<#0106>aggravating, as hot spot causes rotor to blow in direction to produce harder contact, and packing and rotor grow into each other.	Vibration tends to build up in level at constant speed; may be accomplished by mechanical squeal noise.	Reduce speed as necessary to reduce vibration to tolerable levels and to clear rub. Straighten rotor as discussed in paragraphs 231-3.11.2 through 231-3.11.2.6.

231-3.11.2.5 On single- or multishaft ships where a bowed shaft may be slowed, a bowed rotor may be straightened safely and quickly by carefully using both the ahead and astern throttles to produce a low rotational speed as a result of the differential (ahead versus astern) torque. Under this condition, the bowed rotor is bathed uniformly with sufficient heat to correct the bow quickly. Exercise care to adjust throttles so that speed is just below the vibration point and to ensure that excessive temperatures do not result in differential expansions sufficient to develop a rub.

231-3.11.2.6 Under special circumstances, on a multishaft ship, where ship speed cannot be reduced to that required by the bowed shaft, it is acceptable to use the astern turbine to slow the vibrating shaft. Operating the astern turbine with the rotor turning in the ahead direction does abnormally heat the LP turbine from the counterrotating blades, adding heat to the already hot inlet steam. Prolonged application of astern steam (more than 10 minutes) is not recommended; other shafts should be slowed as much as conditions permit to reduce the quantity of astern steam used. Monitor the differential expansion indicator, if installed, and take the indicated corrective action to avoid rubs. LP turbine heating and cooling characteristics and their effects on clearances are discussed in paragraphs 231-3.5.2.4 through 231-3.5.2.5.3, which should be used for guidance.

SECTION 4

CASUALTY CONTROL AND SAFETY PRECAUTIONS

231-4.1 CASUALTY CONTROL

231-4.1.1 SUPERSEDED BY NSTM 079. The requirements of this section have been superseded by S9086-CN-STM-030, NSTM Chapter 079, Volume 3, Damage Control - Engineering Casualty Control.

231-4.2 SAFETY PRECAUTIONS

231-4.2.1 GENERAL. Steam turbine operators should be well trained in safety precautions related to steam turbines as well as general safety precautions. This section serves to provide background on steam turbine safety concerns. In case of conflict between this and any other safety document, operators should follow authorized safety procedures and then notify NAVSEA by a Technical Manual Deficiency Evaluation Report (TMDER) to reconcile the conflict.

231-4.2.2 FIRE. A Navy steam turbine has steam heated surfaces and is lubricated by flammable lubricating oil per MIL-L-17331 (Military Symbol 2190TEP). This oil's flash point, the lowest temperature at which vapors can be ignited by a small flame, is 400°F. Its ignition temperature, the lowest temperature at which a heated substance catches fire in air and continues to burn, is 650°F. Saturated steam temperatures can be between 400°F and 500°F and superheated steam temperatures are higher and depend on the plant design. The potential for an engine room fire exists, but the danger can be kept low if the following precautions are observed.

231-4.2.2.1 All steam heated surfaces should be lagged with thermal insulating lagging, during operation except when steam joint inspection requires temporary removal of lagging pads. Lagging helps to prevent thermal stresses in the turbine casing by allowing more gradual thermal transients across turbine surfaces, but it also can absorb leaking oil, creating possible fire hazard. External lagging surfaces should be painted, not just to resist traffic damage, but also to reduce the potential for absorbing leaking oil. Lagged areas near oil components or mechanical joints may be shielded with sheet metal.

231-4.2.2.2 Lagging should be inspected often for oil films or signs of oil-soaked lagging, which should be reported immediately. Remove oil-soaked lagging from steam surfaces and contain or repair the oil leak as soon as possible.

231-4.2.2.3 Oil system mechanical joints may have male or female flanges or similar flange designs to decrease the chance of radial spraying of oil on steam heated surfaces in case of gasket failure. Sheet metal shields may be installed between oil system mechanical joints and hot turbine surfaces.

231-4.2.2.4 Propulsion turbine or turbine generator oil seals or oil deflectors which are not operating properly can leak oil or even mist oil into the engine room. Keep open flames away and take action to correct the seals or deflectors.

231-4.2.2.5 Steam turbines in operation need not necessarily be secured due to minor oil leaks. Report these to the officer in charge for evaluation and direction.

231-4.2.2.6 Ships with steam turbines should not have oil leaks at the completion of an availability. Leaks should be repaired, not just to reduce the housekeeping burdens for the crew, but also to reduce the risk of fire in the engine room.

231-4.2.2.7 There has been one reported Navy shipboard fire from oil leaking or spraying on steam heated surfaces, and oil soaked lagging has caused considerable smoke in engine rooms. The risk of fire still remains. Take all precautions.

231-4.2.3 OVERSPEED. As a general rule, each Navy turbine is designed to remain intact at 120 percent of the nominal design continuous full power RPM, normally listed in the turbine technical manual. In general, speed limiters are used on submarine main propulsion steam turbines and overspeed trip devices are used on SSTG turbines to prevent uncontrolled overspeeding. They are normally set to actuate at 110 plus or minus 2 percent of the full power RPM listed in the turbine technical manual. See paragraphs 231-2.3.7 to 231-2.3.11 and 231-8.20.6 for more information. The only known cause of catastrophic turbine failure, with blades flying through the turbine casing, has been uncontrolled overspeed incidents. No lives are known to have been lost during these incidents, but the potential for death or injury is obviously there. The chance of having an overspeed casualty is reduced if the following precautions are observed.

231-4.2.3.1 Do not operate a steam turbine if overspeed protective devices are not operating properly. There have been cases where SSTG overspeed trips would not trip to stop the turbine, but operation was continued and the turbine oversped and flew apart. Sometimes blades and coupling are thrown across the room. Usually the turbine rotor breaks into three parts and the casing hogs up and opens. Besides causing the fear of death or injury in personnel, this causes the time consuming delays and expense of major repairs. If a turbine, especially an SSTG turbine, must be run without properly operating overspeed protective devices, verify proper operation of the manual trip and take these extraordinary precautions:

a. Station a dedicated operator at the manual trip for the SSTG trip throttle valve.

b. Limit the steam allowed to the turbine by manually throttling down a steam valve if possible.

c. Troubleshoot and correct the problem as soon as possible, with technical assistance. See paragraph 231-6.1.3.

231-4.2.3.2 When checking speed trip devices, use at least two different means of measuring RPM. See paragraph 231-2.3.11 - Tachometer.

231-4.2.3.3 Test overspeed devices at the required periodicity, typically annually. Do not overtest and wear out these devices.

231-4.2.3.4 Overspeed devices should be repaired, reassembled or reset only by qualified mechanics. See paragraph 231-6.1.3, Technical Assistance.

231-4.2.3.5 Parts used in overspeed devices are critical parts and should only be procured from qualified suppliers. See paragraph 231-6.17.8.

231-4.2.4 ROTATING MACHINERY. See paragraphs 231-3.2.1, 231-3.2.4, and 231-5.7 for precautions and instructions before rotating a propulsion or SSTG turbine.

SECTION 5

OPERATING LIMITS AND PRECAUTIONS

231-5.1 ESTABLISHED OPERATING LIMITS AND PRECAUTIONS

231-5.1.1 GENERAL. The main propulsion turbine, lineshaft equipment, and ship service turbine generator (SSTG) turbine may possibly be subjected to certain conditions that will weaken parts but will not necessarily cause immediate failure. Limits described are established to avoid this degradation, which could cause a crippling failure at some inopportune time.

231-5.1.2 GENERAL LIMITS. The limits generally intended to avoid accumulation of stress cycles that produce premature parts failure are:

a. Restricting power to the technical manual full-power rating

b. Restricting steady-state torque to that corresponding to full-power rating

c. Limiting temperature to values specified for various turbine sections

d. Limiting degree and number of overpressures and overtemperatures in steam chests

e. Driving all shafts except in emergency operation

f. Maintaining equal RPM during steady-state operation

g. Avoiding quick starts with improper warm-ups

231-5.1.3 SPECIFIC LIMITS. While the limits discussed in paragraph 231-5.1.2 are generally applicable to all turbines, in-service experience with or characteristics of certain designs may have made specific additional restrictions necessary. These additional restrictions are included as specific limitations in the applicable equipment technical manual, or affected ships were notified of them by official correspondence. Unless otherwise indicated in this section, the equipment technical manual, or official trial data, maintain equal RPM on all driving shafts when other shaft(s) are locked or trailed during emergencies.

231-5.1.3.1 The operating limits specified in paragraph 231-5.1.3 and in the equipment technical manual apply to normal or special operating conditions and circumstances. Under unusual or abnormal conditions and circumstances not covered in paragraph 231-5.1.3, the safety of ship and crew and mission accomplishment are overriding considerations that fall within the responsibility of the Commanding Officer.

231-5.2 GENERAL LIMITS UNDER ADVERSE CONDITIONS

231-5.2.1 GENERAL. Situations can develop, because of damage or other adverse conditions, that will raise the question as to the maximum ship speed (propeller turns) that can be made safely. Two examples of such situations are towing a disabled ship and remaining underway with the hull in a condition of abnormal displacement that may occur from flooding. Both of these situations change the normal relation of shaft horsepower-to-propeller turns, and each requires higher power and torque at all speeds.

231-5.2.2 EXCESSIVE TORQUE. The potential danger in these situations is not that excessive powers can be developed (since this can be controlled by not firing the boilers or otherwise generating steam over the full-power rating) but that propulsion system torque can be considerably greater than that corresponding to full power. This is explained by the fact that steam turbine power is essentially proportional to steam flow, and if, for a given condition of steam flow, turbine speed is halved, torque will approximately double.

231-5.2.3 GEARED-TURBINE DRIVE. For a geared-turbine drive, this proportional relation, plus the fact that first-stage turbine pressure is approximately proportional to steam flow, can be used to estimate approximately the speed that will cause full-power torque to be developed. The relation can be expressed by

$$\frac{Pe}{Ne} = \frac{Pd}{Nd}$$

Where:

Pe = first-stage shell pressure under the emergency situation (or steam chest pressure with certain hand-nozzle control valves open), in psig

Ne = propeller shaft RPM under the emergency situation

Pd = design first-stage pressure corresponding to full power under normal conditions (or steam chest pressure corresponding to full power with the same hand-nozzle control valves open as for Pe), in psig

Nd = design full-power propeller RPM

231-5.2.4 PRESSURE RATIOS. Pressure ratios for the emergency condition shall be equal to, or less than, the design ratio to avoid overload torques. The same relation is presented by the curve shown in Figure 231-5-1. It must be emphasized that operating at off-design condition (such as shown in curve 0A) increases, among other things, the possibility of blade resonances in the operating range becoming critical, thereby increasing the chances of fatigue failure. It is therefore prudent to use the slowest speed commensurate with the existing situation.

231-5.3 OPERATING WITH A PROPELLER SHAFT WINDMILLING

231-5.3.1 GENERAL. Under circumstances where a turbine- and gear-driven propeller shaft cannot be powered but is otherwise in a condition that the shaft can be safely rotated, allowing the shaft to windmill is preferable to locking. Windmilling produces less drag on the ship and therefore minimizes the additional power other engine(s) are required to deliver at any ship speed.

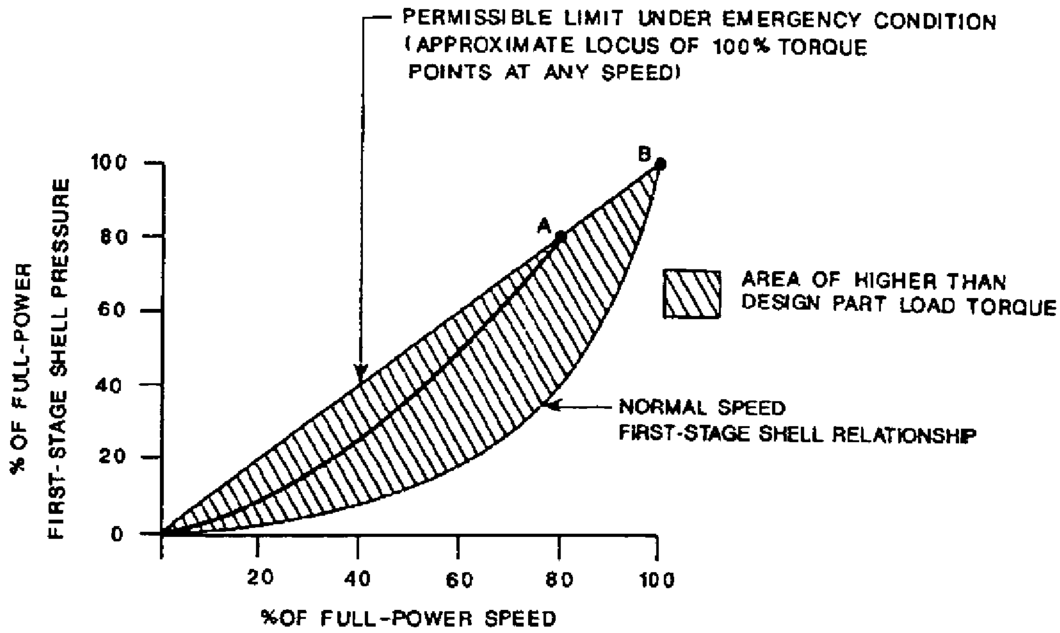


Figure 231-5-1 Pressure Ratio Curve

231-5.3.2 TURBINE HEATING. Windmilling does, however, produce abnormal turbine heating (paragraph 231-3.5.2.2). The maximum speed that can be maintained on a windmilling shaft is that speed for which heat in from friction and heat out to various cooler media equalize at maximum permissible internal temperatures. Of course, reducing heat in or increasing the amount of heat dissipated will reduce internal temperatures. Heat in is reduced by lowering speed and by reducing density of media inside the turbine. Heat dissipation can be increased by introducing cooling media into the system or otherwise improving heat loss from turbine.

231-5.3.3 PRECAUTIONS. Practical means of limiting temperatures and general precautions to be taken when a shaft is windmilling are:

a. Reduce speed (designs are generally based on not exceeding 40 percent of full-power shaft speed on the windmilling shaft).

b. Maintain proper lubrication.

c. Maintain lowest practical main condenser pressure (best vacuum).

231-5.4 OPERATING WITH A PROPELLER SHAFT LOCKED

231-5.4.1 EMERGENCY SITUATION. In an emergency situation, ships may operate with one shaft (or two shafts for a four-shaft ship) locked by the turning gear.

231-5.4.2 PROPELLER SHAFT LIMITS. Propeller shaft RPM limits in Table 231-5-1 are based on ship trial data and should not be exceeded. Precautions that should also be used are listed in paragraphs 231-5.2 through 231-5.2.4 and 231-5.5.2 through 231-5.6.3.1.

Table 231-5-1 LIMITING SPEEDS FOR LOCKED OR TRAILED SHAFTS

Ship Class	Propeller RPM of Driving Shaft(s) with One (or Two) Shafts Locked	Propeller RPM of Driving Shaft(s) with One (or Two) Shafts Trailing
AGF-3	195	200
AOE-1	113	119
AOR-1	87	91
CG-16	250	275
CG-26	205	225
CV-59-64, 66-67	140 (130)	155 (145)
DDG-2/20	230	250
DDG-37	240	255
LHA-1	150	155
LPD-1	178	192
LPD-4	180	180
LSD-36	180	180

231-5.5 OPERATING WITH TURBINE DISCONNECTED

231-5.5.1 PROPULSION TURBINE TYPES. Single-screw ships with two casing turbines have emergency provisions for driving by either turbine (paragraphs 231-3.7 through 231-3.7.4). In a cross-compound high- and low-pressure (HP and LP) turbine arrangement, the singled-up condition requires special pipe sections, orifices, desuperheaters, depressurizers, valves, and blank flanges. This equipment is supplied to the ship, with instructions for its use in the technical manual. The turbine being bypassed can be windmilled by the operating turbine through the gear but most probably will be disconnected by removing the coupling between the idle turbine and the gear. The mode of operation will usually be dictated by the reason for disconnecting one turbine.

231-5.5.2 TWO SINGLE-CASING TURBINES. The arrangement of two single casings driving into one reduction gear and exhausting steam to separate condensers is common in certain ships. In these ships, the singling-up procedure is considerably simplified since redirecting of main steam flow is not required and coupling designs are such that a turbine can be disconnected in a few minutes.

231-5.6 LIMITS AND PRECAUTIONS

231-5.6.1 GENERAL. Exercise prudent control of locked- and trailed-shaft operations to preserve the life of the propulsion turbines (231-5.4 through 231-5.5.2).

a. Operation with other than all driving shafts shall be limited to emergency situations. This does not preclude locking and unlocking shaft drills considered necessary for casualty training exercises. If a shaft becomes disabled, it is preferable that it be disconnected and trailed rather than locked, if conditions permit.

b. When steaming on a steady course and speed (not maneuvering or changing course), maintain equal RPM on all driving shafts when shafts are locked or trailing, as well as when all shafts are driving, unless existing full-power trial requirements state otherwise.

c. For shafts with permanently installed torsionmeters, do not exceed maximum full-power ahead torque on all driving shafts when other shafts are locked or trailing.

231-5.6.2 CROSS-COMPOUND ARRANGEMENT. The following limitations and precautions are applicable to cross-compound turbines operating singly:

a. Operating the HP turbine alone prohibits astern turbine operation.

b. Torque-speed relation will differ from that of two turbines driving, and less than full-power steam flow must be used to limit turbine to full-power torque. (Use speed or other parameter limit specified by manufacturer. If information is unavailable, see paragraphs 231-5.2 through 231-5.2.4 for guidance.)

c. If fitted with desuperheater, steam is to be desuperheated to 750°F or lower before admission to the LP turbine.

d. In using the desuperheater, careful regulation is necessary to avoid introducing slugs of water into turbine.

e. If a turbine is disconnected while hot, circulate lubricating oil to the turbine, if possible, until temperature rise across bearing is 15°F or less.

f. When a turbine is allowed to windmill, observe thermometers or hot spot thermocouples and keep temperatures within stated limits.

231-5.6.3 TWO SINGLE-TURBINE ARRANGEMENTS. The following limitations and precautions are applicable to the two single-casing drives because of their special features:

a. The temperature in the turbine exhaust is limited to a relatively low value by a flexible exhaust element. In a windmilling condition, sprays shall be turned on and vacuum held at the best level possible. Monitor temperature in exhaust and increase speed consistent with driving turbine first-stage pressure, or speed limit, and hood temperature. Failure to maintain exhaust temperature within recommended range may subject the turbine materials to temperatures in excess of their design values and decrease allowable life. Also, excessive temperature will damage nonmetallic connections between turbine and condenser, causing possible rupture or vacuum loss.

b. Disconnect ahead and astern valve reach rods from a disengaged turbine if the hydraulic system control oil is unavailable. (Driving turbine is to be operated in override range of throttle.)

c. Monitor turbine internal temperature indicators for signs of overheating.

d. Use turbine cooling sprays.

e. If vacuum cannot be maintained, permit air to circulate by opening manhole or inspection covers in each turbine casing. Keep condenser circulating water flowing, if practical.

f. Remove turbine lagging, if warranted.

231-5.6.3.1 Windmilling one or more shafts tends to overload other engines. Limitations on driving turbines in the technical manual and paragraphs 231-5.1 through 231-5.1.3.1 apply. Lack of lubrication will require that the windmilling shaft be locked.

231-5.7 DOCKSIDE OPERATION

231-5.7.1 SPECIFIC RESTRICTIONS. During dockside operation or testing at low astern RPM, it is possible to expose the LP turbine to excessive temperatures that can result in serious turbine rubs. To prevent turbine damage from extended operation at low astern RPM, the following restrictions should be used for all ships with uncontrolled superheat:

a. Ahead operation may be any RPM; there is no limitation.

b. Astern operation above 20 percent of ahead full-power propeller may be any RPM; there is no limitation.

c. Astern operation below 20 percent of ahead full-power propeller RPM shall be limited to 30 minutes of continuous operation or 1 hour of total accumulated operation when cycling ahead and astern for equal periods of time.

NOTE

A minimum of 14 hours of cooldown time shall be allowed before additional astern operation.

231-5.7.2 FULL RESTRICTIONS. The limits imposed by items a through c (paragraph 231-5.7.1) are intended to be used in addition to those already specified by the turbine manufacturer and the equipment manual, and imposed by the dockside operating environment.

231-5.8 OTHER CONSIDERATIONS

231-5.8.1 VACUUM. Economical operation of condensing turbines depends on the vacuum maintained; therefore, keep packing and joints in condition to prevent leaks. A high vacuum once attained can be maintained easily and permanently by taking the following precautions:

a. Keep gland packing in good condition, and keep a feather of steam coming from steam-packed glands at all times.

b. Keep air removal equipment in excellent condition.

231-5.8.2 EXHAUST CASING PROTECTION AGAINST EXCESSIVE PRESSURE. To protect the turbine exhaust casing against excessive pressure, different devices are fitted on different turbines. These devices include sentinel valves, full relief valves, spring-loaded exhaust valves, steam and exhaust valve interlocks, and back-pressure trips. Unless otherwise specified on applicable drawings, the casing relief valve on turbine generators should be set at 10 psig and the back-pressure trip set at 5 psig.

231-5.8.2.1 Relief Valves and Back Pressure Trips. All condensing SSTG turbines now in use should have a relief valve large enough to prevent the pressure in the condenser from exceeding 30 psig with the throttle open or a back-pressure trip.

231-5.8.2.2 Sentinel Valves. Small relief valves on turbines are installed as sentinel valves to warn of an undue increase in back pressure. They are not designed to relieve the turbine completely of excessive pressures, and when a sentinel valve pops, immediately ascertain the cause of the pressure increase. If necessary, shut down the turbine.

231-5.8.2.2.1 Relief valves are sometimes provided with a shroud to permit piping away the steam if the valve blows. The piping leading away from this valve shall not be plugged, nor shall any valve be installed in this line.

231-5.8.2.3 Exhaust Line Relief Valves Precautions. Where stop valves are installed in the turbine exhaust lines and full-relief valves are not provided on the casing or on the exhaust line between the turbine and the first stop valve, the turbine trip throttle valve handwheel should be marked, preferably with a red warning disk underneath the handwheel, DON'T OPEN WHILE EXHAUST VALVE IS CLOSED. Similarly, the exhaust valve handwheel should be marked DON'T CLOSE WHILE TRIP THROTTLE VALVE IS OPEN. In addition to the warning disks, which should be installed at once, the situation should be immediately brought to the attention of the Naval Sea Systems Command so that steps can be taken to provide proper relief valves and spring-loaded exhaust valves.

SECTION 6

STEAM TURBINE MAINTENANCE

231-6.1 GENERAL

231-6.1.1 MAINTENANCE REQUIREMENT CARDS. Preventive maintenance information in this section complements the Maintenance Requirement Cards (MRC), the basic job order requirement for each task supplied for programmed preventive maintenance under the maintenance and material management (3-M) system. MRC's for propulsion and ship service turbine generator (SSTG) turbines are required for each preventive maintenance task. Where the Planned Maintenance System (PMS) coverage applies, preventive maintenance should be conducted in accordance with the MRC's. For any turbine work, however, use the documents in the overhaul work package. In the absence of specific documents, use the manufacturers' technical manuals or detailed drawings.

231-6.1.1.1 The information in this section, the manufacturers' equipment manual, ship information books, and other applicable documents provide detailed procedures for conducting corrective maintenance and for evaluating the results of required inspections and tests to an extent not covered by the MRC's.

231-6.1.2 MAINTAINING TURBINE RELIABILITY. Steam turbine construction is relatively simple, and its basic reliability stems from this fact. The turbine contains few moving parts and practically no wearing-type parts. In fact, if various linkages are kept greased, lubricating oil and steam are kept scrupulously clean, and proper warming and drying procedures are followed, the turbine can be expected to operate for 30 years without replacing major parts. Primary shipboard maintenance objectives include a continuing awareness of possible changes in performance and therefore consist of keeping systems clean and periodically checking and verifying adequacy of internal clearances. Consult the turbine manufacturer's closing report for the actual as-shipped turbine internal clearances.

231-6.1.3 TECHNICAL ASSISTANCE. Technical assistance is available to any ship having equipment problems including turbine problems, or to any intermediate level (intermediate maintenance activity (IMA) and tender) or depot level (shipyard) repair facility needing turbine maintenance support.

231-6.1.3.1 Direct Fleet Support. Turbine maintenance beyond the capability of the ship's force (organizational level) is normally delayed until an availability at a tender (intermediate level). Technical assistance for turbine maintenance at this level is provided by the Direct Fleet Support (DFS) Program described in NAVSEAINST 4350.6, Direct Fleet Support Program; Policies and Procedures For. Under DFS, the Type Commander can request technical assistance from the Fleet Technical Support Center, Atlantic (FTSCLANT) or Pacific (FTSCPAC). A visit will be made if necessary to resolve the equipment problem, provide fleet on-the-job training, and assess available logistic support from technical manuals, allowance parts lists (APL), and similar sources. If necessary the support centers can arrange for additional technical assistance from the Carderock Division Naval Surface Warfare Center (CDNSWC), Philadelphia, or the turbine manufacturer. Direct Fleet Support provides troubleshooting and guidance to prevent problems from reoccurring. To actually repair a turbine or turbine component will require ship's force corrective maintenance, or, if more involved, tender or shipyard availability.

231-6.1.3.2 Factory Assistance. All turbine original equipment manufacturers (OEM) have company field engineers available to provide turbine technical assistance to shipyards and through the support centers to the fleet. Hiring field engineers has proved to be highly cost effective, especially during a major turbine overhaul or a very technical repair. Field engineers can also provide factory support, including drawings, parts, machining, and factory engineering support. Some repair activities have found in-factory turbine overhauls more cost effective than in-yard overhauls. Central OEM service numbers and other useful phone numbers are listed below:

a. Demag Delaval Turbomachinery Corp.
   P. O. Box 8788
   Trenton, NJ 08650
   609-890-5158

b. Dresser-Rand Steam Turbine Div.
   (Formerly GE Turbines)
   37 Coats Street
   P. O. Box 592
   Wellsville, NY 14895-0592
   716-593-1234

c. Northrop Grumman Marine Systems
   (Formerly Westinghouse Turbines)
   401 E. Hendy Avenue
   P. O. Box 3499
   Sunnyvale, CA 94088
   408-735-2639

d. Kingsbury Bearings
   10385 Drummond Road
   Philadelphia, PA 19154
   215-824-4000

e. Waukesha Bearings
   P.O. Box 1616
   Waukesha, WI 53187-1616
   414-547-3381

f. Pioneer Bearing Co.
   116 Beacon St.
   South San Francisco, CA 94080-6988
   415-871-8144

g. Woodward Governor Co.
   P.O. Box 3800
   Loveland, CO 80539-3800
   303-663-3950

h. In Place Machining
   1929 N. Buffum St.
   Milwaukee, WI 53212
   800-833-3575

231-6.1.3.3 Qualified Personnel. Only qualified personnel shall disassemble, reassemble or repair steam turbines or steam turbine components or manufacture turbine parts. Qualified personnel are personnel with a history of successful performance of the required action. This history shall be documented, demonstrated and part of an established reputation that is auditable. A qualified organization has and uses qualified personnel to perform the required action(s). A qualified manufacturer of steam turbine parts has an established reputation of success of supplying the required parts to all drawing dimensions, finishes, parameters, processes and material specifications. These definitions and requirements do not supersede any existing qualification requirements.

231-6.1.3.4 Naval Sea Systems Command Assistance. NAVSEA in Washington, D.C. is not meant to assist in normal steam turbine maintenance problems. NAVSEA does need to be notified of turbine problems that might directly relate to turbine design deficiencies, quality deficiencies, and life cycle management. Examples of these types of problems are given throughout this chapter including:

a. Blade failures
b. Casing cracks
c. Turbine welding
d. Overspeed destruction
e. Opening turbine casings
f. Journal grooving
g. Bearing heavy wipes
h. Major steam leaks
i. Repetitive failures
j. Main turbine balancing
k. Material substitution needed

231-6.2 STEAM-PATH CLEARANCES

231-6.2.1 GENERAL. The efficient and successful operation of a propulsion turbine depends largely on maintaining the design steam-path clearances. The technical manual clearance diagram shows clearances that are a compromise between the thermodynamicist's ideal of zero clearance and the mechanical designer's estimate of the clearances that can be maintained without serious rubs. Study the clearance diagram and note the following information:

a. Critical dimensions are specified in decimal dimensions to a thousandth of an inch (mil).
b. In any impulse stage rotating row, upstream clearances are generally smaller than downstream clearances.
c. Nominally, axial clearances become larger proportionately with distance from the thrust bearing.
d. Nominally, radial clearances are proportional to the radii of the seal surfaces.
e. Radial sealing is not required in pure impulse stages but is almost always provided in partial and full reaction stages.
f. Close clearance steam-path parts are usually shaped to permit contact and wear without generating excessive heat.

g. Allowable wear in thrust and journal bearings is limited to that which will not make internal clearances critical.

231-6.2.2 CLEARANCES OUT OF SPECIFICATION. Clearances not in agreement with the design clearances given in the manufacturer's technical manual do not necessarily warrant repositioning the rotor. Contact NAVSEA for evaluation of conditions and action to be taken.

231-6.2.2.1 The manufacturer's closing report shows as-shipped clearances. This report is given to the ship during construction to be filed permanently aboard. Some of the clearances may be out of drawing specifications, but they have been analyzed and approved by cognizant engineers.

231-6.2.2.2 Wheel clearances, blade to stationary part, may vary by up to 10 mils between inboard and outboard sides of a turbine stage.

231-6.2.2.3 It is important that a total rotor float can be taken without rubs or the possibility of rubs due to thermal movements. Rubs shall be avoided, even at the expense of efficiency.

231-6.2.3 IMPORTANCE OF AXIAL CLEARANCES. To some extent, reaction- and impulse-type blading will show a decrease in nozzle-blade efficiency as the axial distance between the stationary element and rotating row is increased. This is the primary reason for designing axial clearances to practical minimums although additional benefits accrue from reduced rotor lengths and weights. Although it is desirable to minimize axial clearances from an efficiency standpoint, clearances must remain adequate to allow necessary differential casing-to-rotor growths without serious rubs. Reasons for these differential expansions and discussion of the rather significant transient changes associated with astern operation are discussed in paragraphs 231-3.2.2.2 and 231-3.5.2.5 through 231-3.5.2.5.3. The various designs consider these factors, and proper turbine operation requires that clearances be maintained close to those shown in the manufacturer's clearance diagram. The means provided for maintaining and checking axial rotor position, both in operation and when idle, as well as permissible variations in axial clearances are discussed in paragraphs 231-6.2.5 through 231-6.2.5.4.

231-6.2.3.1 Thrust Bearing Clearances. Thrust bearing clearances are axial clearances taken with the thrust bearing installed. They are normally taken with the rotor cold (after a 6-hour minimum cooldown) for consistency in readings. In case of a suspected casualty, however, take thrust readings as soon as possible after the rotor shaft is stopped and lube oil is secured, and record the temperatures. The thrust collar, and therefore the rotor, is moved between the forward and aft thrust bearing surfaces (Figure 231-2-10).

a. End Play Cold Clearance. The thrust bearing end play or cold clearance is the distance the thrust collar can be moved between the forward and aft thrust bearing faces without applying appreciable load to either face. This end play clearance is provided to allow for oil flow, to establish an oil film, and to allow for thermal expansion of the thrust bearing components. The required end play clearance can be found in the turbine technical manual, but typical end play with the machine cold is 0.008 to 0.014 inch. This end play is set when the rotor is installed by placing filler plates (shims) behind each thrust bearing. One filler plate is machined to set the required nozzle block clearance between the stationary nozzle block and the rotor's first-stage blades. The other filler plate is machined to set the thrust bearing end play as required. After loaded turbine operation, the thrust bearing parts take a permanent set called bedding-in. Bedding-in can add 0.001- to 0.003-inch clearance to each side of the thrust bearing. A cold clearance end play can therefore have 2 to 6 mils added to it, and typical end play could become 0.008 to 0.020 inch.

b. Running Clearance. The thrust bearing running clearance is taken hot with the turbine running and usually taken with a rotor position indicator. The running clearance includes the end play, amount of bedding-in, elastic deflection of parts under load, and wear. The technical manual will give the maximum running clearance for a thrust bearing. A typical maximum running clearance could be 0.030 inch.

CAUTION

Take extreme care when checking total float clearance. Internal parts could be damaged by contact with each other.

231-6.2.3.2 Total Float. The total float clearance is the axial clearance taken with the thrust bearing shoes removed. Do not check total float clearance without referring to the technical manual or without manufacturer's permission. Always take extreme care as internal parts can easily be damaged. To check total float clearance, remove the thrust bearing shoes and cautiously jack the rotor forward and aft measuring the full travel. A typical measurement is 0.200 inch. Total float can be compared with previous values when looking for possible internal damage and displaced parts.

231-6.2.4 IMPORTANCE OF RADIAL CLEARANCES. Wear in gland seals, diaphragm packing, and blading seals (which can all be grouped as a radial type) causes steam to bypass the active steam-path blade area and degrades turbine performance. Maintaining radial position of the rotor to the tolerance specified by the manufacturer, through permissible wear of the journal bearings, is important in minimizing packing and seal wear. Radial clearances should not be large enough to affect tooth contact patterns of connected gears or to misaline the coupling. A detailed discussion of the effect on performance of variation of radial clearances is presented in paragraphs 231-3.10.1 through 231-3.10.1.4.

231-6.2.5 MEASUREMENT TECHNIQUES. Certain measurements are required to establish that the turbine rotor position in the casing is within prescribed limits. The means are normally provided to measure only axial rotor position with the turbine in operation. Radial position measurements require turbine shutdown. Devices designed to give information on rotor position during operation do not directly measure any specific clearance in the turbine but indicate a motion relative to a zero setting. That zero setting should correspond to and be set with the turbine cold and the rotor verified to be properly positioned on the basis of direct internal clearance measurement. Rotor position measurements, however, may exceed drawing measurements or tolerances. If the rotor position is out of tolerance refer to paragraph 231-6.2.1. Position the rotor to obtain optimum thrust bearing clearance or float. If necessary, local waiver and analysis are available. The various types of indicators are discussed in paragraphs 231-6.2.5.1 through 231-6.2.5.3.

231-6.2.5.1 Rotor Position Indicators. Turbines have two types of axial rotor position indicators. The first is a dial indicator that contacts the turbine shaft end nearest the thrust bearing. It monitors the position of the thrust collar in the thrust bearing and therefore provides a measure of thrust position and condition. The second is the differential expansion indicator, which is mounted at and contacts the rotor at the end away from the thrust bearing. It is normally a dial or pointer-and-scale indicator, and its purpose is to measure the relative position of the rotor and casing during heating transients or maneuvering conditions. Differential indicators, although in place on many older units, are no longer installed on newer units and may have been removed on many older units. Manufacturers' differential expansion clearances, once proved on initial units, do not require monitoring. Each of the indicators has the limits of safe operation either stamped on the device itself or specified in the technical manual. A change in the reading past the technical manual limit would indicate babbitt loss from the thrust shoes. Where limits are unspecified, a loss of half the minimum axial clearance may be cause for concern. Parts of these indicators that contact the moving shaft are susceptible to wear. Since measurement is normally in mils, a small amount of wear may be significant in interpreting the reading. Settings of these devices should be checked periodically; wear can be minimized by a light touch in use.

231-6.2.5.2 Finger Gages. Finger gages, or finger pieces, (used in some turbines of older design) give a ready visual check on the forward and aft position of the turbine rotor. A finger gage is a less precise measure of rotor position than a dial indicator. In most cases the finger piece consists of a plate or pointer secured to the turbine with an indicator extended to the proximity of the shaft. Three lines are usually scribed on the plate to show the normal and maximum positions. In other arrangements the indicator extends into a shaft groove that permits feeler readings to be taken between the indicator and shaft.

231-6.2.5.3 Taper Gages. Although dial-type rotor position indicators and finger gages give a quick check of rotor position, they are not to be used arbitrarily as a basis for replacing thrust shoes or for repositioning the rotor. If suspected to be wrong, verify the position of the rotor by measuring the nozzle-to-blade or other clearance as specified on the MRC or by the turbine manufacturer. Access is usually provided through casing openings fitted with removable covers, and a taper gage is provided as a special tool for measuring the clearance. In use, most taper gages are coated with a light film of Prussian blue on the tapered surface and inserted into the space between the stationary element and the moving blade to the depth permitted by the opening. If used properly, the result will be a sharp line of demarcation between the gage section where the blue has been scraped clean and where the blue is undisturbed. The demarcation line can be read directly, where the gage is marked, and interpolation between marks made as necessary to read to the nearest thousandth of an inch. Outside micrometers can be used for independently measuring the gage thickness at the demarcation line. Note that an accurate measurement does not automatically result. Some taper gages do not use bluing. Consult the turbine technical manual for specific gages. The gage is intended to be a precision measuring instrument, and care shall be exercised to ensure that the gage is inserted in a direction perpendicular to the surface contacted and that it seats firmly in the access opening. Repeat measurements until consistent readings are obtained. Precede the measurement by positioning the rotor hard forward or aft in the thrust bearing, as specified in the applicable MRC or manufacturer's technical manual. Direct comparison with the design clearance can then be made. A variation of more than 10 percent should be cause for further investigation. If the rotor position needs adjusting, rotate the rotor by hand or turning gear to be sure there is no rubbing after adjusting the clearance.

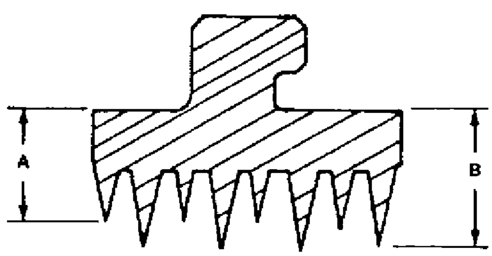


Figure 231-6-1 Gland Packing Ring

231-6.2.5.4 Crown Thickness for Radial Seals. The effect of increased packing clearance on performance and normal wear criteria are discussed in paragraphs 231-3.10.1 through 231-3.10.1.4. Seal wear is not uniform around the circumference and since most casing distortions result from vertical temperature gradients, wear is usually more extensive in the bottom of the lower half (and the top of the upper half) of shaft and gland packing. The normal direct measurement of seal-to-shaft clearance at the casing horizontal joint, with the upper half gland or casing cover removed, will therefore not be representative of the overall packing condition. When visual inspection indicates that wear is extensive and a quantitative measure is required to determine required repair or other corrective action, crown thickness measurements can be taken at sufficient radial positions to establish the wear pattern. The thickness reading may be converted to wear by comparison with the design value, an unworn part of the packing, or a spare packing. Crown thickness measurements of a gland packing ring are shown in Figure 231-6-1.

231-6.3 THRUST BEARINGS

231-6.3.1 MEASURING CLEARANCES. When properly alined and lubricated, the thrust bearing should be capable of unlimited operation without significant wear (paragraphs 231-3.10.3.3 through 231-3.10.3.4.7). The small tolerance for change in thrust clearance (resulting from tight axial blade clearance), however, makes it prudent to check the thrust periodically to verify that the clearance has not become excessive. Rotor position indicators permit rapid checking and are intended to establish that no untoward motion of the rotor has occurred (paragraphs 231-6.2.5.1 and 231-6.2.5.2). These devices may be worn or may lack precision, so rotor position and clearances should be verified by moving the rotor through its thrust clearance and measuring motion with a separately fitted dial indicator capable of measuring to a thousandth of an inch. With all newer turbines special jacking tools are provided to move the rotor axially and are sized so that excessive forces will not be applied routinely. Take care to bar or jack rotor in accordance with procedures given in the equipment manual. The following procedure may be used as guidance:

1. The turbine shall be at ambient temperature with throttles shut and lube oil not circulated for 24 hours (this prevents film in thrust bearing clearance). For an SSTG set, the generator rotor may remain attached.
2. The thrust bearing shall be fully assembled, with the as-found shims, and the thrust cover and end plate bolted on tightly.
3. The dial indicator is to be located so that the jack force will not cause local rotor bending and erroneously indicate motions greater than clearance. If possible, the indicator should be installed at the end of the rotor opposite to the applied force.
4. If lube oil was supplied between 24 and 48 hours, the timing gear may be jogged on then off to free up the rotor.
5. Keeping the rotor continuously moving within the thrust clearance, jack the rotor one way until it is firmly against the thrust shoes. Expect to use considerable force to move the rotor, especially when trying to break it away from rest. With a small positioning force on the jack that shows no noticeable motion on the indicator for a slight increase, zero the dial indicator.
6. Shift the rotor through its clearance in the opposite direction, repeating the clearance measuring procedure.
7. The indicated reading is the thrust clearance. Repeat the evolution several times to prove repeatability and to establish confidence in the reading. A reading over specification indicates thrust shoe wear. A reading under specification indicates incomplete travel within the thrust bearing clearance and may indicate thrust bearing parts out of location.

231-6.3.2 ADJUSTING THRUST BEARING OIL CLEARANCE. The manufacturers' technical manual, detail drawing, or MRC's specify design oil clearance at some nominal value with a tolerance and a limiting value. If, as a result of a maintenance check or other action, the thrust clearance is determined to be above specified limits, but thrust shoes and other thrust parts are otherwise in good condition, the clearance can be readjusted. This is done by replacing the shim (provided for this purpose) with another that is properly flat and dimensionally the same, except for thickness. The thickness is adjusted to make up the difference between the as-found and design values. The thickness at any section of the replacement shim should not be allowed to vary more than 0.0005 inch. The re-establishing of the design clearance (within tolerance) should be verified by moving the rotor through the thrust clearance after reassembly with the new shim or replacement of parts.

231-6.3.3 ADJUSTING ROTOR AXIAL POSITION. Provisions are incorporated in the turbine thrust bearing for axially repositioning the rotor should it become necessary. Such repositioning should be required only as a result of casing dimensional changes or, more likely, after a rotor changeout. These changes will be disclosed by the nozzle or other blading axial clearances deviating from design valves. The radial clearance between the rotor blade shroud and the tightening rings (tip seals) cannot be adjusted after the tightening rings are machined and installed. After the rotor is set axially, the radial clearance between the shroud and tightening rings may deviate from design. A 10 percent variation in these axial and radial clearances will normally be of no concern; indications of larger deviations should be closely watched for accurate measurement (axial clearance paragraphs 231-6.2.3 and 231-6.2.3.1). Wheel clearances, blade to stationary part, may vary by up to 10 mils between the inboard and the outboard sides of a turbine. If possible, determine the axial clearance at other locations to verify that rotor repositioning is warranted. After clearance measurements are determined to be accurate, the rotor can be repositioned by adjusting the shims used to position the thrust housing in the turbine casing.

231-6.3.3.1 Shimming. Where two shims set the position, one shim will be increased in thickness by the amount of the desired shift and the other shim will be decreased by the same amount. Variation in thickness at any section of a replacement shim should not be more than 0.0005 inch. The re-establishment of the desired blading clearance should be checked after complete reassembly of the thrust bearing or replacement of parts. Do not add shims to an existing shim'manufacture a new shim.

231-6.3.4 INSPECTING THRUST BEARINGS. Thrust bearing components (see paragraph 231-2.9) should be inspected to drawing dimensions. Thrust bearing pads should be inspected for wear and imperfections using Table 231-6-1 as a guide. In addition, thrust bearing pads occasionally show a ″sunburst″ pattern on the babbitt surface. This can be caused by cooling too slowly in the factory or by extensive thermal cycling in service. If this pattern is a faint etching with no depth to the sunburst ″rays,″ then it is acceptable for service. If babbitt is missing at the rays, it is unacceptable and should be discarded. Faulty thrust shoes are not rebabbitted but replaced (see paragraph 231-2.9.2.2).

231-6.4 JOURNAL BEARINGS

231-6.4.1 BEARING ALINEMENT. Bearings will be seated either spherically or cylindrically, depending on the length of bearing and intended service. The bottom half of the bearing is normally supported by a structure integral with the bottom half of the casing. A bearing cap encloses the top half of the bearing and, because the bearing cap is bolted on, provides access to the bearing.

Table 231-6-1 BEARING INSPECTION TERMS

Term	Description or Action
Smear	Babbitt has moved. Affected area is less than 1/4 of bearing length and less than 1/4 of the way around the circumference. Usually a local high spot. No change in bearing clearance. A good bearing.
Light wipe	Babbitt has moved. More than a smear. No major change in bearing clearance (within clearance specs). A good bearing. May require some dressing or scraping.
Wipe	Greater than a light wipe. Slightly exceeds bearing clearance specs. Replace bearing except in an emergency. Reuse after scraping.
Heavy wipe	Greater than a wipe. Exceeds bearing clearance specs. Not salvageable. Replace and send for rebabbitting.
Total wipe	Worst wipe. Almost all babbitt is gone. Shaft probably severely damaged. Replace bearing. Investigate cause to prevent recurrence.

Table 231-6-1 BEARING INSPECTION TERMS - Continued

Term	Description or Action
Pounded	Babbitt is severely cracked, loose, or missing. Babbitt may be peened over. Replace bearing. Investigate cause to prevent recurrence.
Resistance temperature detector (RTD) Resistance temperature element (RTE) Exposed	Babbitt is missing over RTD tip. Replace bearing. Faulty installation of RTD.
RTD dimpled	Babbitt is lowered over RTD, but intact. Small circular dimple may show minor cracks. Bearing may be reused. Causes no known symptoms.
Misalined	Babbitt shows an edge-to-center triangular pattern (misalined top to bottom) or diagonal pattern (misalined port to starboard). May be reused if light wipe. Should be replaced at repair facility. Aline correctly.
Porosity	Babbitt shows random pits from air bubbles at manufacturer. May be reused if 90 percent good surface and pits are tiny. Should be replaced at repair facility.
Sponginess	Babbitt has a soft, porous, mottled finish. Manufacturing problem. Replace bearing at once since it has lost its bearing properties.
Electrostatic pitting	Babbitt shows pits grouped together, usually near the loaded area in the lower half. Replace bearing and investigate cause.

231-6.4.1.1 Cylindrically Seated. Cylindrically seated bearings generally have cylindrical seats that roll into and match a cylindrical fit in the bearing housing or pedestal. A cylindrically seated turbine journal bearing is shown in Figure 231-6-2.

231-6.4.1.2 Spherically Seated Bearings. Spherically seated bearings, which are common in propulsion turbines, are characterized by spherical surfaces at the outside of the bearing shell and the inside of the bearing housing. Bearing spherical surfaces may be machined into the shell itself or into removable pads that also permit shimming of the bearing to adjust rotor height or athwartship position.

231-6.4.1.2.1 The mating spherical seat may be machined in the bearing housing or may be machined in a bearing support bushing, which in turn fits a cylindrical bore in the bearing housing. A spherically seated bearing in a mating bushing is shown in Figure 231-6-3. A spherically seated bearing is shown in Figure 231-6-4.

231-6.4.1.2.2 The advantage of a spherically seated bearing is that the bearing can be shifted at installation into perfect alinement with the journal. Whether the bearing shifts to accommodate transient misalignment in operation is open to question. Major turbine manufacturers (on the basis of their particular experience) design fits between the bearing shell and housing that vary from 0.006 inch interference, to size-to-size, to 0.012 inch loose. The heavily pinched bearings certainly will not move, but there is some evidence that clearance bearings shift to accommodate the slow-acting motions associated with casing thermal distortions. Consult the turbine technical manual to determine this fit.

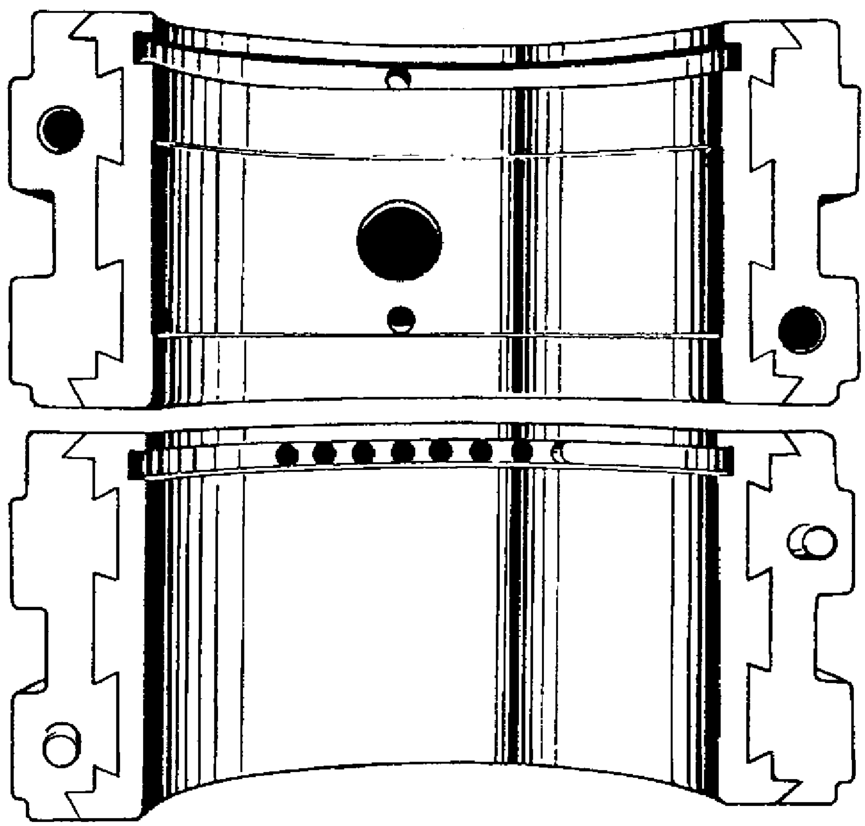


Figure 231-6-2 Cylindrically Seated Turbine Journal Bearing

231-6.4.1.3 Alining Spherically Seated Bearings. When replacing a spherically seated journal bearing, examine the ball of the bearing and the ball seat of the bearing bracket to make sure there are no burrs or imperfections that will interfere with the seating of the bearing. Dress the surfaces, if necessary, and make sure they are clean. Wipe the bearing and journal surfaces with lubricating oil. The procedure for alining the bearing (also see paragraph 231-6.4.3.4) is as follows:

1. Roll the lower-half bearing into position, and aline it in the ball seat of the bearing bracket so measurements taken at A, B, C, and D (Figure 231-6-5) are equal.
2. Back off the jack screws of the rotor lifting device so that the shaft rests in the lower-half bearing. Positioning the lower-half bearing results in lining up the bearing in one direction only. To determine whether bearing is tilted fore and aft in its ball seat, make a lead wire check.
3. Lay a piece of soft lead wire or plastic gage material along the journal surface between X and Y (Figure 231-6-5). The wire should not be more than 0.010 inch greater in diameter than the bearing clearance. Using a larger diameter wire may cause excessive force to be applied and damage the babbitt.
4. Bolt the top half of the bearing in place, remove it, and measure the compressed thickness of the wire. If there is a variation in thickness at X and Y, it will indicate how much and in what direction the bearing is tilted axially. If bearing is tilted, again lift the shaft and correct the tilt.
5. To determine bearing cap pinch fit, refer to paragraph 231-6.4.4.
6. A useful tool for alining bearings is blueing the journal and rolling it through the bearing lower half. Lightly scraping the bearing babbitt can produce a satisfactory contact pattern (see Figure 231-6-6). Blue contact

should be centered in the lower 30 to 40 degrees of the bearing over 90 percent of the bearing axial length.

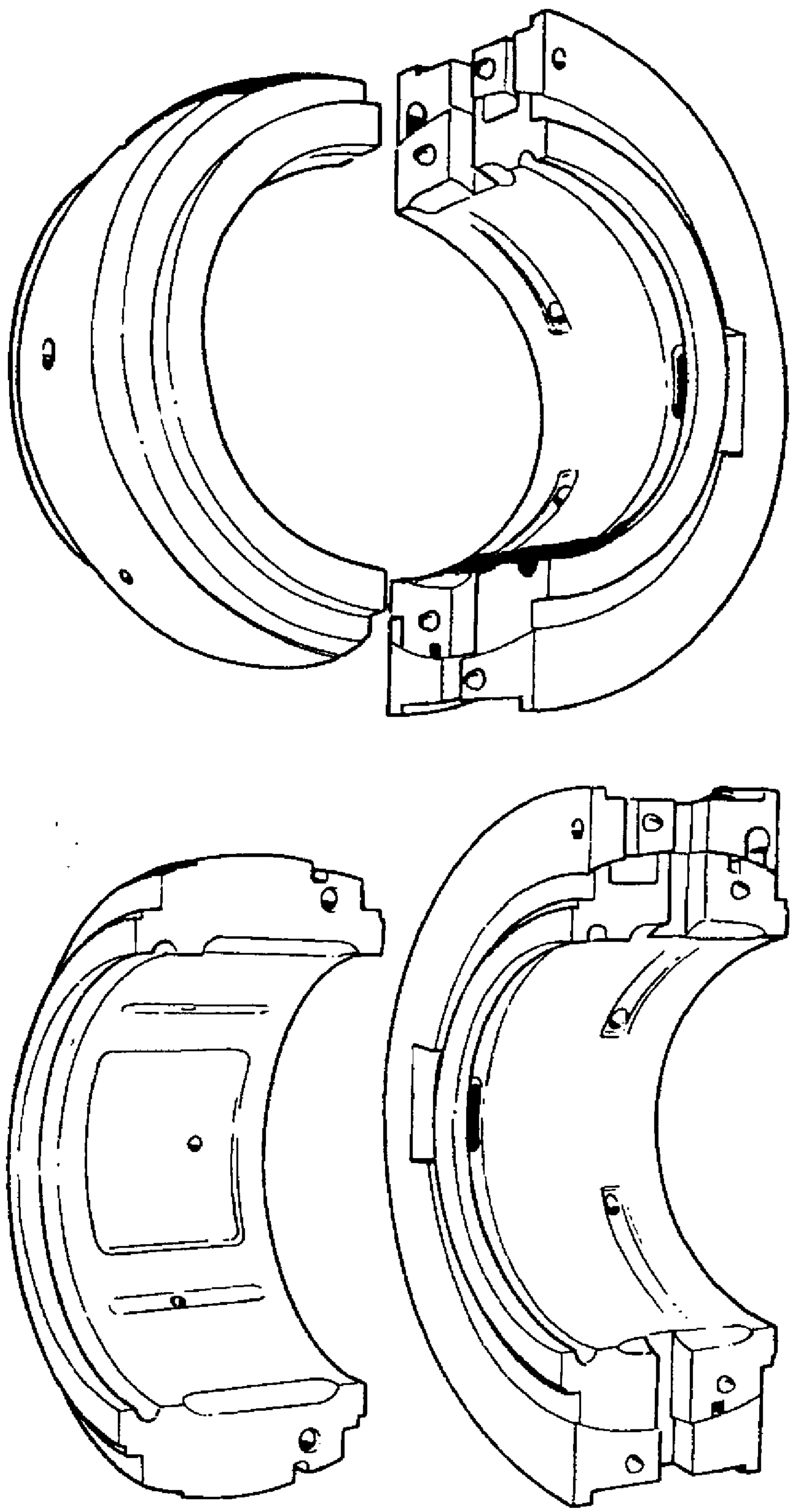


Figure 231-6-3 Spherically Seated Bearing in a Cylindrically Seated Bushing

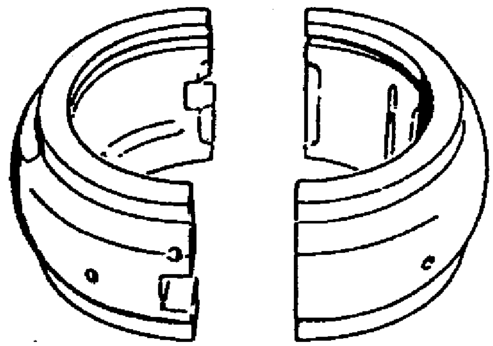


Figure 231-6-4 Spherically Seated Bearing Shell

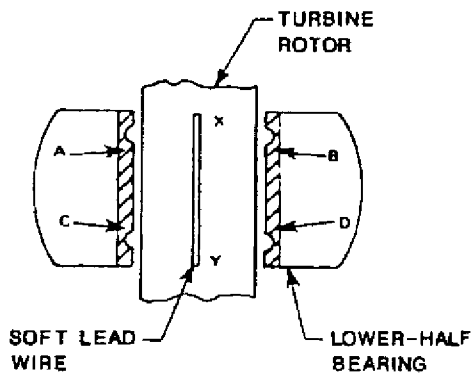


Figure 231-6-5 Bearing Alinement Checking Method

231-6.4.2 INSPECTING JOURNAL BEARINGS. Journal bearings need not be disassembled periodically for inspection. Journal bearing disassembly and inspection is recommended at each complete overhaul. If, however, a PMS MRC depth micrometer reading discloses an out-of-tolerance measurement or if an operational difficulty develops, disassembly and inspection may be necessary. Confirm MRC readings before disassembly and inspection. When deciding whether to open a bearing consider the following questions:

a. Is there an availability in the near future?

b. Are there babbitt flakes in the strainers or sight flow indicators?

c. Has this bearing shown a high temperature?

d. Has there been a rapid drop in depth micrometer readings?

e. Has there been a recent loss of lube oil to the bearing?

f. Does the depth micrometer reading exceed the stamped constant plus the allowable wear? If called for, make a detailed visual examination of the various bearing surfaces and take appropriate wear measurements. To remove journal bearings, remove the upper bearing bracket, and use the turbine rotor jack tools to jack the rotor just enough to roll the lower bearing half. Jacking too high will crush the upper labyrinth seals. Unsolder any RTD wires. Always set the rotor on dummy bearings (correct size for the journal) while the turbine journal bearings are removed.

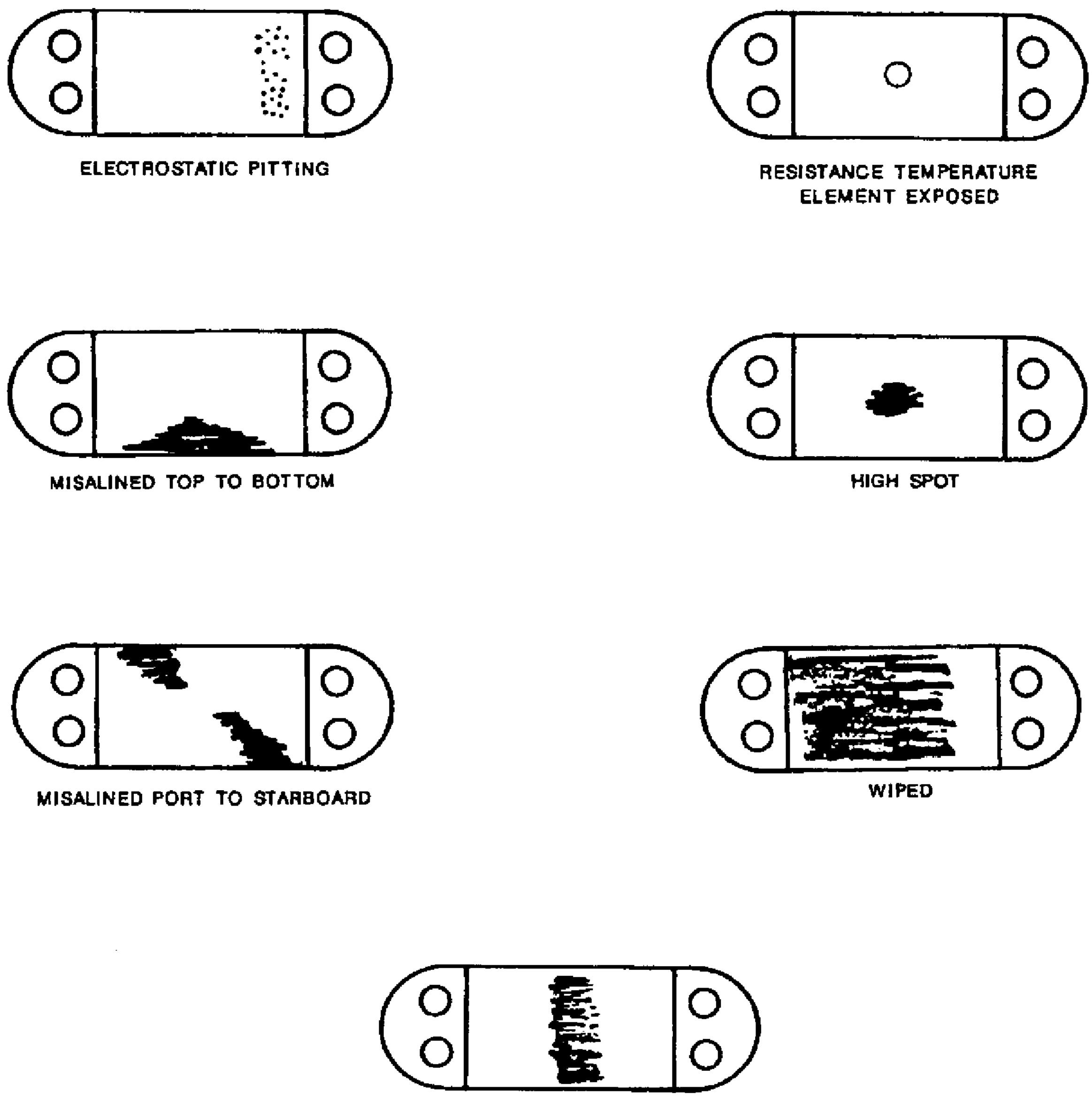


Figure 231-6-6 Bearing Patterns

231-6.4.2.1 SSTG Bearing Alinement. If a steam turbine journal bearing needs to be replaced or disassembled for inspection, rolling out of the bearing from the bearing housing may affect the SSTG alinement. The SSTG alinement is set by the design of the installation, which may or may not include a reduction gear. To maintain alinement in some installations, the turbine forward bearing (low-pressure (LP) bearing, refer to manufacturer's technical manual) is elevated above the high-pressure (HP) bearing and the two pinion bearings to compensate for the sag in the turbine shaft and the differential in expansion of the bearing supports, and to ensure uniform pinion bearing loading when the SSTG is operating. For these reasons consult the manufacturer's drawings and instruction manual for bearing alinement data when replacing or installing the journal bearings. Either steam turbine journal bearing may be replaced, but the gear pinion bearings should be replaced together.

231-6.4.2.2 Wear Pattern and Wear Limits. A properly installed bearing that has been in service for some time will usually have a worn or polished area centered in the lower half of the bearing. This is caused by the journal contacting the bearing surfaces during startup, shutdown, and operation of the jacking gear. The width of the worn area will vary, depending on the degree of hand fitting, if any, previously performed and on the amount of wear. Variations in pattern width in the fore-and-aft direction are a measure of angular alinement between the bearing and journal under stopped or very low-speed conditions. The contact pattern does not normally represent the running position of the journal, since contact should be eliminated completely by the hydrodynamic film developed and the minimum film location will be displaced angularly from the bottom of the bearing by an amount set by the speed and load conditions.

231-6.4.2.2.1 Bearing wear would be almost completely eliminated under ideal conditions. Some journal-to-bearing rubbing contact, however, is made on each start, and oil contaminants accelerate wear. Most turbine technical manuals specify the wear limit at which the bearing is to be rebabbitted or replaced. When this information is unavailable in the manual or detail drawings, a rule of thumb is that the wear limit equals twice the minimum bearing clearance. Common bearing wear patterns are shown in Figure 231-6-6. Bearing inspection terms and recommended actions are listed in Table 231-6-1.

231-6.4.2.2.2 Inspection of bearing contact is required any time a fixed arc bearing is replaced. A satisfactory bearing fit is indicated by a blue contact centered in the lower 30 to 40 degrees of the bearing over 90 percent of the bearing length.

231-6.4.2.3 Discoloration of Babbitt. Discoloration of bearing babbitt surfaces almost always indicates lubrication problems. Moisture and iron oxide in the oil and operation under relatively high temperatures can produce a tin oxide coating on the bearing that may vary from a mottled dark gray to highly polished black. Tin oxide can also result from contact with water containing chlorides or from contamination by desiccants containing chlorides. Chlorides can result when a lube oil cooler leak allows seawater to enter the lube oil system. The coating is very hard and builds up to reduce bearing clearance. The babbitt does not have the softness of the original babbitt, and if coating particles break off or the oil system is contaminated, journal scoring is likely. Thin coatings can normally be satisfactorily scraped without exceeding clearance limits, but thick coatings usually require bearing replacement or rebabbitting. Eliminating water leaks, proper oil cooldown procedures, and maximum use of oil purifiers will minimize difficulties. For more information on tin oxide, see NSTM Chapter 243, Propulsion Shafting.

231-6.4.2.3.1 High temperature can create decomposition deposits (hydrocarbons) on bearing surfaces, journals, or drain passages. The color can vary from tan to dark brown, to black, depending on the degree of overheating and oil decomposition. If discoloration cannot be traced to overheating, investigate the possibility that oil additives are plating out. A chemical laboratory qualified to analyze lubricating oils should check the oil.

231-6.4.2.4 Bonding. Improper bonding of the babbitt to the bearing shell is no longer a common occurrence. Faulty bonding usually results from deviation from proper babbitting procedures and inadequate quality assurance inspection techniques, rather than the age of the bearing or abuse in service. If a bond is faulty at an outside babbitt-to-shell juncture, the improper bond can be detected visually or by squeezing oil from the interface using thumb pressure applied at the babbitt adjacent to the area to be checked. One hundred percent bonding is not required for satisfactory operation, and minor bonding flaws that meet the acceptance standard should not be cause for replacement. An in-service bearing bond is satisfactory if no single edge is completely unbonded at the babbitt-to-shell interface. If any cracks between the babbitt and the shell run a whole side (arc or horizontal joint of bearing), then the bearing should be either replaced or further evaluated by ultrasonic testing (UT) per DOD-STD-2183, Bond Testing, Babbitt-Lined Bearings. Liquid penetrant testing (dye penetrant testing) may be used to clarify visible cracks, but it should not be used as an acceptance test. Also, do not use UT alone as an in-service bond acceptance test. Too strict a test will result in satisfactorily bonded bearings being replaced unnecessarily. A bearing that has failed the visual test may be reinstalled until replacement is more convenient, as long as the bearing babbitt remains intact.

231-6.4.3 CLEARANCE AND WEAR MEASUREMENTS. Clearance and wear measurement techniques are illustrated in Figure 231-6-7 and Figure 231-6-8 measurement actions and additional information relevant to steam turbines are listed in Table 231-6-2. Bearing oil film can affect depth micrometer readings, bridge gage readings, and the taking of leads. Secure oil supply to the bearing 24 hours before measurement, if practical. Procedures necessary to perform the actions listed in Table 231-6-2 are given below. Also see NSTM Chapter 243 and NSTM Chapter 244, Propulsion Bearings and Seals.

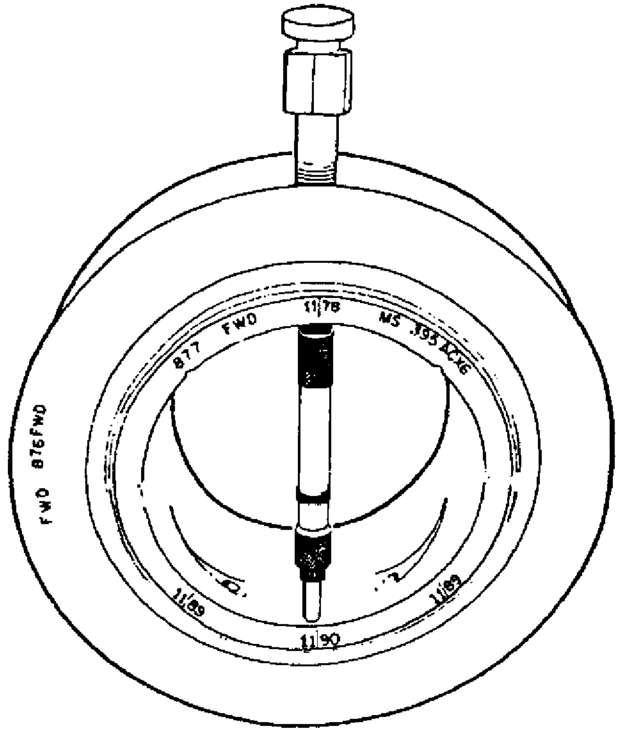
231-6.4.3.1 General. Bearing wear readings should be carefully taken at regular intervals and a permanent record of bearing clearances kept in the ship's log and attached label plate. Also, whenever a bearing cap is disturbed, a new set of depth micrometer readings is required. These readings are intended as a check of operating conditions between overhauls. See paragraph 231-7.2.

231-6.4.3.2 Measurements By Crown Thickness. Whenever bearing temperatures, poor gear tooth contact, or unsatisfactory operation is believed to be caused by worn bearings, measure the wear by the crown-thickness method. Micrometer depth gage or bridge gage readings alone are insufficient.

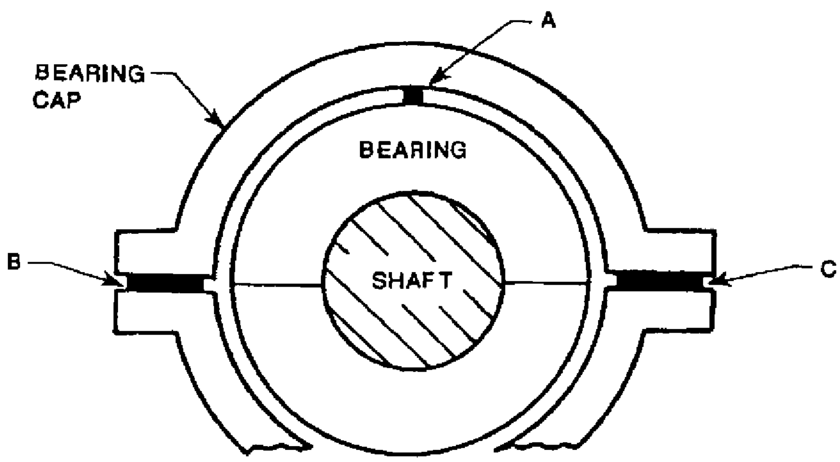
231-6.4.3.3 Alternate Measurement Methods. If bridge gages are specified but are unavailable, a special alternate method of measuring rotor concentricity and bearing wear can be used (paragraph 231-6.4.3.3.1).

231-6.4.3.3.1 Bearings are to be rebabbitted by ships or field activities only in an emergency. Ships and field activities are to check stock for replacement bearings before rebabbitting (paragraph 231-8.14.2). If rebabbitting a bearing becomes necessary, ensure that the shaft, when supported in the rebabbitted bearing, lies in the same position it originally occupied. Take micrometer readings at each end of the turbine between the rotor and the machined surface of the gland cap. Using a depth micrometer, measure at three locations: top dead-center vertical; in line with the first bolt, which secures removable gland packing; and above horizontal on each side of top dead-center.

231-6.4.3.3.2 These readings will indicate the concentricity of the rotor within the gland bore. If all three readings are identical, the rotor and gland are concentric. The distances measured are those between the rotor and the under side of the gland cap overhang. If no overhang exists, remove gland packing and measure from the rotor to the gland cap. Take the measurements approximately 1/4 inch from the end of the gland cap; do not measure from any inclined rotor surface. If necessary to obtain a flat measuring surface, replace parts of gland cap, but not gland packings. After removal, the HP and LP turbine bearings can be measured for wear by checking the crown thickness. Using the outside micrometer, check the crown thickness of the bearing at three specific longitudinal planes on the end of the bearing. Radial marks scribed on the bearing end designate the measuring planes. The value of bearing thickness dimensions at the time of manufacture is stamped on the end of each bearing. If no scribe lines exist, they can be established when a new bearing is installed. The lower bearing half shall have three lines scribed on both ends of the bearing shell: one at the geometric centerline and two at a 45-degree angle from the center scribe. The upper half shall have one line scribed on both ends of the bearing shell, 90 degrees from the bearing joint. Measure bearing shell thickness at the locations specified, and stamp these readings on the ends of the shell. After obtaining the three readings, the stamped thickness constant associated with each one is subtracted from the crown thickness reading to give the increase in bearing clearance or amount of bearing wear. When this wear exceeds the allowable wear limits stated in the applicable technical manual, replace the bearing. For more information on babbitt and rebabbitting, see paragraph 231-8.14.



A. WEAR MEASUREMENT



B. CHECKING PINCH FIT OF BEARING CAP

Figure 231-6-7 Journal Bearing Wear Measurements

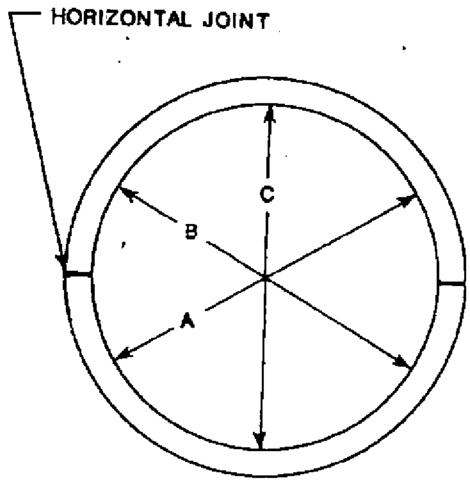


Figure 231-6-8 Measuring Points for Wear of Journal Bearings

Table 231-6-2. MEASUREMENTS AND SIGNIFICANCE*

Action	Remarks
1. Depth micrometer check See paragraph 231-7.2	If excessive weardown, verify need for replacement of bearing by disassembly and performing any of actions 2, 3, 4, or 5. Resolve discrepancies.
2. Shell crown thickness measurement	A deviation of 0.001 to 0.002 inch of measurement from stamped constant can be expected because of variation in temperature and micrometer feel.
3. Bridge gage readings	Use pin gages and templates for checkout of bridge gage, if provided and available.
4. Taking leads	See paragraph 231-6.4.6.3 for check of bearing shell-to-housing fit by leads.
5. OD & ID measurements	Measure ID of bearing at points A, B, and C at both ends and in the middle of the bearing (total nine measurements) per Figure 231-6-8 and OD of journal at multiple locations. Difference is bearing clearance.

NOTE: See NSTM Chapter 244.

231-6.4.3.4 Tilted-Pad Bearing Disassembled Clearance. The tilted-pad (shoe) journal bearing diametral clearance is calculated as the housing ring inside diameter minus 1/2 the sum of the four pad thicknesses minus the journal diameter. To use this method, remove the tilted-pad bearing from the bearing bracket, disassemble the housing ring, remove the tilt pads, and reassemble the housing ring. Measure the inside diameter at three positions with a deep-throated inside micrometer with ball ends. This is to ensure that the horizontal joint of the bearing housing has not spread or pinched and that the joint has not been worked, creating an oval instead of circular ID. Also, inspect the internal surfaces of the bearing housing at the contact points of the pads to verify no wear. Measure the tilt pads at the thickest part of the pad, usually just off the center. Measure the journal and calculate the diametral clearance using the above formula. If the clearance is satisfactory, reassemble and reinstall the bearing and record the clearance. See Figure 231-6-9.

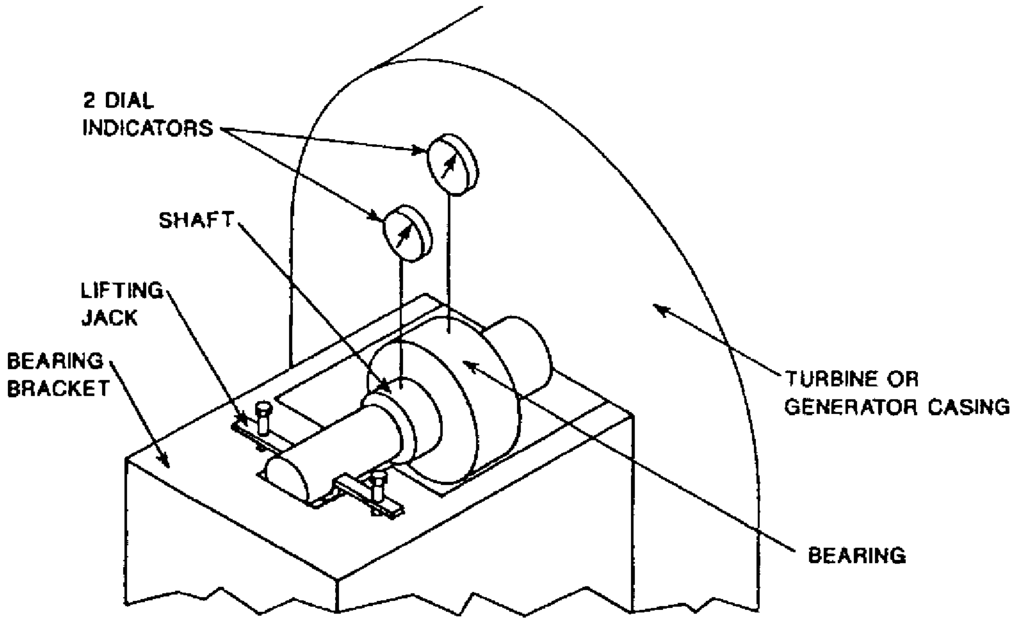


Figure 231-6-9 Bearing Clearance Without Disassembly

231-6.4.3.4.1 For some tilted-pad bearing designs the diametral clearance can be measured without bearing disassembly by shaft lift measurement. Verify that clearances of seals, packings, and other turbine components are large enough to allow the rotor to be lifted in the journal clearance without damage. With the upper bearing bracket up and the tilted-pad bearing still assembled around the journal, install a dial indicator to read journal vertical movement. Jack up the rotor carefully in the bearing clearance, and read the lift of the journal. Use a second dial gage to verify that the bearing is not lifted. Repeat the procedure to show repeatable results. If this shaft lift diametral clearance is repeatable within 0.001 inch of the previously measured disassembled clearance, then this method may be used on this tilted-pad bearing design.

231-6.4.4 BEARING CAP PINCH FIT. A journal bearing is kept in alinement in its bearing bracket by the pinch fit between the ball of the bearing and the ball seat of the bearing bracket. The pinch fit should be according to manufacturer's specifications, typically between 0.001 and 0.005 inch. The method of taking measurements to check the pinch fit is shown in Figure 231-6-7(b). Plastigage or other thickness-measuring material may be used in place of lead wire if necessary. The procedure is as follows:

1. Remove the bearing cap.
2. Lay a piece of soft lead wire on the top of the ball of the bearing, at A, parallel to the shaft direction. Use 0.020-inch-diameter wire. If the bubbler supply hole is at the top of bearing, lead wiring may be put to either side of the hole and both wires later measured.
3. Place 0.015-inch-thick shims on the horizontal joint of the bearing bracket at B and C, cover as much of the joint as possible, particularly the area adjacent to the bolts and assembly, and bolt down the bearing cap. Lead wires may also be placed alongside the shims on the joint to confirm that the joint gap actually equalled the shim thickness and the joint was not being held open by any obstruction. Since the joint of the bearing bracket will be held open 0.015 inch by the shims, the soft lead wire will be compressed to less than 0.015 inch by an amount equal to the pinch fit.
4. Remove the bearing cap and measure the compressed lead wire. If the wire measures between 0.015 and 0.020

inch, there is no pinch fit. If the wire measures less than 0.015 inch, the difference between the measurement and 0.015 is the pinch fit. For example, if the compressed wire measures 0.012 inch, the pinch fit is 0.015 minus 0.012, or 0.003 inch.

5. If the pinch fit is between 0.001 and 0.005 inch, remove the shims and replace the bearing cap.

6. Refer to paragraphs 231-8.6.3 and 231-8.7.3 for instructions on making up the joint and installing fasteners.

231-6.4.4.1 If the measurement taken in step 4 shows that the pinch fit is either less than specified or nonexistent, it cannot be immediately determined, without manufacturing drawings, whether the diameter of the ball seat in the bearing bracket is too large, or the ball of the bearing is too small. The spare bearing should be installed, alined properly, and another pinch fit taken to see if the situation has been corrected. If the check of the pinch fit on the second bearing shows that the pinch fit is still less than required, a temporary arrangement can be made to produce the required pinch fit. Cut a star-shaped shim from a piece of steel shim stock of the thickness necessary to produce the correct pinch fit. If necessary, drill a hole in the star-shaped shim so as not to obstruct oil flow to the bubbler. Lay the shim on the top-center of the bearing and bolt down the bearing cap. The star shim should be temporary, and the original pinch fit should be corrected at the earliest opportunity. If a star shim is installed, a suitable tag should be affixed to the bearing cap noting that the star shim is installed. This will alert anyone removing the bearing cap to prevent the shim from falling into the oil drain.

231-6.4.4.2 If the measurement taken in paragraph 231-6.4.4, step 4 shows that the pinch fit is greater than specified, the upper bearing cap can be shimmed at a shipyard availability. The upper bearing cap joint can be skim-cut to allow for a shim of reasonable thickness. This can be installed with a sealer and countersunk screws. These actions can be done with local engineering direction and without a NAVSEA waiver.

231-6.4.5 REBORING DIRECTIONS. Each bearing that can be rebabbitted and that is not bored concentrically with the outer surface of the shell is provided with a concentric reference shoulder at each end for reboring. Unless otherwise marked on the bearing, the outside diameter of the shell shall be used as the reference shoulder for reboring.

231-6.4.5.1 Offsetting Bearing Bores. Bearing bores are not to be offset to achieve turbine-to-gear alinement, but only if required by a drawing or to obtain rotor-to-casing concentricity.

231-6.4.6 PRECAUTIONS WHEN WORKING ON BEARINGS. It is quite possible to disassemble a bearing that is working perfectly and, through poor work practices, have difficulty in operation following reassembly. Disassemble bearings only with justification because of the dangers of improper reassembly. If disassembly is necessary, it is important that the work be done, or closely supervised, by experienced people. Information on the initial preparation and some of the precautions to be taken during bearing disassembly or reassembly is given in paragraphs 231-6.4.6.1 through 231-6.4.6.1.5. Nonstandard propulsion turbine bearings are available (paragraph 231-8.15.5).

231-6.4.6.1 General. Study the technical manual clearance diagram to determine what motions (axial and radial) are permitted by the internal clearances.

231-6.4.6.1.1 Lifting gear, such as jacks, slings, and chain hoists, must provide positive control over the rotor position when the bearing is no longer supporting the shaft weight. If special tools for this purpose are provided by the turbine manufacturer, they will be detailed in the drawings or technical manual.

231-6.4.6.1.2 A dial indicator can be used in measuring the shaft lift necessary to remove the lower-half bearing. Lift should be limited to that required to free the lower half. Values between 0.005 inch and the radial bearing clearance are suitable. Ensure that lifting devices are set to lift the rotor precisely vertically, since side shift of the rotor will pinch the bearing despite adequate lift. Lifting the rotor slightly more than twice the bearing clearance will cause the bearing to bind after partial roll out. Excessive lifts can damage close-fitting elements.

231-6.4.6.1.3 The fit of the bearing shell in the housing influences journal-to-bearing alinement and performance. It is important, therefore, that the external surface of the bearing shell and the mating interior surface of the bearing bracket not be damaged by mishandling, and that these surfaces are clean and free from burrs in reassembly.

231-6.4.6.1.4 If difficulty occurs in rolling out the bearing lower half, it can be started by placing a zinc slab edgewise on one side of the joint (where the bearing halves meet) and striking the slab with a maul. Similar non-destructive methods can be used.

231-6.4.6.1.5 Make sure that bearings are properly identified when removed to avoid interchanging whole bearings or bearing halves with those of another location.

231-6.4.6.2 Oil-Water Drains. Some steam turbines have oil-water (slop) drains between the outermost shaft seal ring and innermost oil deflector at one or both journal bearings. These chambers have a drain hole in the bottom that can get clogged with lagging and other debris. Ensure that oil-water drains do not get obstructed, so that oil does not get pulled into the turbine. Inspect with a boroscope if necessary and clean as necessary. Verify unobstructed drain flow by pouring a tablespoon of water into the chamber and watching it exit.

231-6.4.6.3 Spherically Seated Bearings. When a spherically seated bearing is reinstalled, the mating spherical surfaces shall be clean and free from burrs and upsets. On size-to-size or clearance spherical fits, oiling the surfaces should permit moving the installed lower-half bearing by hand pressure alone, once the bearing is in place. This freedom of motion will ensure that the weight of the lowered journal will properly aline the lower-half bearing to the journal. On pinch spherical fits, the bearing is not as free to move, and alinement between the journal and the bearing should be verified. The fit of the bearing in its housing can also be checked by using lead wire or plastigage between the bearing shell and cap. The fit can vary from interference to a small clearance. Bearing cap pinch measurement procedures can be found in paragraph 231-6.4.4.

231-6.4.6.4 Alinement Pins. Prior to installing a journal bearing, check relative movement between bearing halves with the bearing half fasteners loose. No movement is acceptable. If movement is detected, check alinement fit in accordance with technical manual or bearing drawing requirements.

231-6.4.6.5 Perform a ″blue″ contact check of the top of the bearing to its seating surface in the bearing bracket. Verify continuous contact around the bubbler supply hole to ensure that oil does not leak between the bearing and the bracket while not adequately supplying the bubbler. Also, after the upper bearing bracket is installed, put a depth micrometer shaft through the bracket down into the bearing to demonstrate that the holes line up and that the depth micrometer will touch the top of the shaft before torquing fasteners.

231-6.4.7 SIGHT FLOW INDICATORS OR BUBBLERS

231-6.4.7.1 Flooded Bubblers. A flooded bubbler is a sight flow indicator that is completely full of lube oil and shows no evidence of oil flow. Bubblers that are completely or partially full of lube oil and show an indication of flow are not flooded bubblers, but acceptable sight flow indicators. Do not stop a turbine, prevent a turbine start, call for emergency repairs, or delay a ship's deployment because of a flooded bubbler alone. The problem is usually with the flow of oil from the bubbler and not with the oil flow to the bearing. Verify that the bearing temperature is satisfactory by bubbler thermometer or resistance temperature element (RTE) temperature, if installed. Lower or raise the turbine speed when possible, and note when the bubbler drains. A bubbler will usually be flooded only in a certain speed range or at zero speed. Notify the watchstanders to monitor bearing temperature, as necessary, in the affected speed range. During the next convenient availability, make the following checks:

a. Disassemble flooded bubbler. Check dimensional correctness of all parts in accordance with design drawings.

b. With indicator disassembled, check for obstructions and alinement of drainage passageways.

c. Check size of flow controlling orifices for correctness.

d. Check bearing and verify that it is in accordance with design drawings.

e. Verify that bearing oil supply pressure and temperature are within design limits.

f. Vent bubbler to check if bearing is pressurized.

231-6.4.7.1.1 Once these checks have been made, continuous unrestricted operation with a flooded bubbler is acceptable.

231-6.4.7.2 Loss of Bubbler Flow. Loss of bubbler flow in the main or ship service turbine generator (SSTG) turbines is acceptable if there are other indications that the bearing has acceptable oil flow. Do not start a turbine without indication of oil flow to all bearings unless flow to the affected bearing is verified by some other means. Some turbines with attached lube oil pumps, which give no flow with the turbine at rest, also have marginal electric auxiliary pumps that will not produce bubbler indications at all bearings. These turbines may be routinely started on the basis of previously unobstructed flow, no recent maintenance on the lube oil system (a source of rags, debris, and other obstructions), and immediate flow indications from the attached pump. Turbines may be rotated up to several revolutions, at turning gear speed or less, with the lube oil system off if residual oil is in the bearing. Pour 2190 TEP oil down the bubbler standpipes if necessary. Do not stop a rotating turbine, call for emergency repairs, or delay a ship's deployment because of loss of bubbler flow alone. The problem is usually with the flow to the bubbler and not with the bearing oil flow. Verify satisfactory bubbler thermometer or RTE temperature. Lower or raise the turbine speed, when possible, and see if flow indications return. Notify watchstander to monitor the bearing temperature as necessary in the affected speed range. During the next convenient availability, make the following checks:

a. Confirm that pressure control valve is satisfactory to give design pressure at bearing pressure gage and design temperature.

b. Check bearing to ensure hole in bearing lines up with passageway to sight flow glass.

c. Check that bearing details in way of feed passage (that is, depth of pressure wedge) are in accordance with design requirement on drawings.

d. Blow out or clean passageways to remove any possible restrictions.

e. Check sight flow glass thermometer bulbs to ensure they are correct size. There should be clearance between bulb and tube for oil to flow up tube and through sight flow glass.

f. Remove sight glass and ensure that tube is securely seated in housing and not allowing oil to flow directly back to drain instead of through the sight flow glass.

231-6.4.7.3 Bearing Temperature Instruments. Thermometers sense this bubbler oil temperature, and the thermometer readings, when properly interpreted, provide a measure of bearing condition during operation. The bearings may also be fitted with RTE's that sense either the bubbler oil temperature or, being embedded in the babbitt, the babbitt temperature near the bearing load line. An RTE is a temperature-measuring instrument in which electrical resistance is used to measure temperature. For this reason RTE's for bearings should not have spliced leads since splicing may change the resistance of the RTE. If RTE wires must be spliced, a qualified technician shall certify the joint as mechanically and electrically sound and recalibrate the RTE assembly. An RTE may be calibrated by verifying its temperature resistance characteristics at two or three ambient temperatures found locally (i.e., 60°F, 70°F, and 80°F). If these resistances comply with Table I of MIL-T-22051, Temperature Element, Resistance, Bearing Babbitt, or ship's electrical maintenance instructions, then the RTE will be calibrated throughout its operating range. Do not endanger RTE's or bearings by attempting to reach the high and low temperatures of Table I of MIL-T-22051.

231-6.4.7.3.1 For more information on sight flow indicators, see paragraph 231-2.11.

231-6.5 WHY BEARINGS FAIL

231-6.5.1 GENERAL. The most frequent causes of naval steam turbine bearing failures are improper installation, excessive dirt, and an inadequate oil supply (film), causing actual rubbing of the journal or thrust runner on the bearing surface. Potential causes of bearing failures are discussed in paragraphs 231-6.5.2 to 231-6.5.5.2.

231-6.5.2 SCORING, WIPING, AND SEIZURE. Wiping usually results from rubbing a shaft on a soft bearing material, such as babbitt, when there is a temporary break in the oil film. Such a condition may be encountered during initial startup of a machine in which there is some misalinement, bearing pinching, insufficient clearance, or other geometrical defect in the bearing or journal. Wiping of bearing material, particularly babbitt, generally provides sufficient clearance or conformity so that further satisfactory operation will be possible. Slow run-in periods are generally desirable so that wiping action may take place without generating extremely high temperatures that could destroy the bearing.

231-6.5.2.1 Babbitt starts to melt or become putty-like at about 465°F and will become completely liquid at 670°F (Grade 2) or 790°F (Grade 3).

231-6.5.2.2 If harder bearing materials were used, such as bronze or aluminum, failure of the oil film or abrasive dirt in the oil would generally result in scoring of both the bearing surface and the journal. Relatively hard particles torn from the bearing surface by initial contact promote scoring and scratching of surfaces, rather than the simple wiping encountered with babbitt.

231-6.5.3 WEAR. Various factors can contribute to bearing wear, but at a much slower rate than that encountered in the wiping and scoring previously discussed. Gradual mechanical wear occurs in any bearing in which the journal and bearing surfaces are not separated by a lubricant film or oil wedge. Starting and stopping, high loads, very low viscosity oils, high operating temperatures, and vibration may contribute to mechanical wear.

231-6.5.3.1 Wear may also be a mild abrasion in which dirt particles passing through the bearing slowly remove a portion of the surface. Similarly, corrosion by contaminants in the oil or by oxidized oil itself may cause wear. Wear under these conditions may be minimized by the proper selection of bearing materials and designs. The effect of dirt particles can best be avoided by designing the bearing to run with a large clearance, by using grooves that continually flush dirt out of the bearing, and by using a bearing material that will readily embed hard particles. Corrosive wear may be avoided by using bearing material that will not be attacked by oil or oil contaminants.

231-6.5.4 ELECTROSTATIC WEAR. Destructive current of about 1 milliampere or more will flow through most journal bearings if the voltage potential difference from the journal to the bearing is approximately 1 volt or higher. Sparks pass through the oil film and cause microscopic pits in the surface of the journal and the bearing. This results in a continual removal of metal from the surface of the bearing and in severe cases from the journal. Trouble may be noticed, however, in a few hours to several years, depending on the magnitude of the current flow.

231-6.5.4.1 Electrical wear shows up as a frosted surface in the loaded zone of the bearing where the oil film is thinnest. Under a microscope the frosted area appears as a cluster of small craters. Each crater results from the fusing temperature generated by the passage of an electrical spark. See paragraph 231-8.15.1.

231-6.5.4.2 Changes in bearing design, bearing materials, or lubricants are ineffective in avoiding the wear from electrical current. There are, however, three methods of preventing current from flowing through the oil film:

a. The bearing may be insulated from its housing by an insulated lining, such as phenolic plastic.
b. A grounding brush may be applied to the shaft to provide a lower resistance path for the current to pass around the bearing.
c. Most effective, although frequently difficult to accomplish with an installed unit, is eliminating the cause of the electrical potential across the bearing.

231-6.5.4.3 Analyzing the electrical system associated with the unit in which the bearing is located and considering possible sources of stray currents or of static charges building up on the turbine rotor may lead the way to eliminating the voltage source.

231-6.5.5 ABRASION. Circumferential scratches on bearing surfaces and embedded hard particles are characteristic of abrasion. The nature of this scratching may vary from an almost microscopic effect with very fine particles, which results in slow wear, to severe scoring, which may be encountered with large, abrasive dirt particles.

231-6.5.5.1 Abrasion may differ from scoring or simple wear by the presence of foreign particles in the bearing surface material. Careful probing with a pick, microscopic examination, and chemical spot testing are all valuable tools in identifying the abrasive material. Once identification has been made, the source of the foreign material may frequently be removed. If this proves impossible, the dirt should be excluded as completely as possible by carefully filtering the oil coming into the bearing and by sealing the bearing housing to prevent the entry of airborne matter.

231-6.5.5.2 The effects of abrasive material may be minimized by using:

a. A hard shaft and as soft a bearing material as possible to increase the ability to absorb foreign particles
b. A large oil film thickness between the journal and bearing with a high viscosity oil and lightly loaded bearing surface

c. Grooves in the bearing to provide maximum flushing action of solid particles from the bearing and to prevent collection at dams and in oil grooves

231-6.6 JOURNALS

231-6.6.1 DESCRIPTION. A turbine bearing journal is the portion of the turbine rotor that rotates inside the journal bearing assembly. The term packing journal is sometimes used for the portion of the rotor that rotates inside the packing assembly. The bearing journal is machined to precise specifications for diameter, roundness, concentricity with other rotor surfaces, surface finish, and taper. The journal is machined from the rotor base material, from a shrunk-on sleeve, or from a layer of chrome plating. The static and dynamic load of the rotor is transferred to the journal bearings at the journals. Undesirable noise and vibration are also transferred to the bearings at the journals.

231-6.6.2 INSPECTION. Because journal condition is so important to bearing wear, noise transmission, and general turbine operation, inspect the journal whenever the bearings are removed. Clean hands before inspecting since much of the inspection is done by feel and foreign substances can cause print marks and damage to journals. Journal inspection terms and recommended actions are listed in Table 231-6-3.

231-6.6.2.1 Additional information on bearing journals can be found in paragraph 231-8.15.

Table 231-6-3 JOURNAL INSPECTION TERMS

Term	Description or Action
Discoloration	Shiny or dull circumferential lines, bands, or blemishes. These have no depth. Take no action.
Phonographing	Phonographic finish of many light scratches caused by fine dirt. Usually over entire area of journal. Take no action if these are not causing bearing babbitt wear.
Scored	Coarser than phonographing. Caused by dirt particles up to 1/32 inch in diameter. If localized, caused by problem area on bearing surface. Dress high spots on journal under factory direction. Clean the oil or investigate the bearing.
Grooved	Coarser than scored. Caused by dirt particles up to 1/8 inch in diameter or problem area on bearing surface. Hone or grind in place or replace the rotor.
Steel (wire) wooling	Similar to grooved. Caused by dirt in babbitt riding on 12 percent chrome alloy journal or thrust collar. Bits of journal alloy in the bearing babbitt can act as a machine tool, which can cause grooving of the rotor. Replace rotor.
Pits	Small spherical holes in journal surface. Take no action if these are not causing bearing babbitt wear or affecting the noise of a noise-controlled turbine.

231-6.7 OIL DEFLECTORS

231-6.7.1 The oil deflectors are made of a soft material (aluminum or bronze) that can sustain light rubs without scoring the shaft. Whenever the bearings are open, the deflectors are accessible and should be inspected and checked for clearance. Verify that the various drains are open and free running. Pay particular attention to the oil and water drain described in 231-2.8.1. If rolling out is necessary, the deflectors may be tapped lightly with a brass or zinc rod at the parting face. If required, drill and tap the ends of the deflector to assist in removal. Also, a segment of an old deflector (remove teeth) may be used to work the deflector around and out. Replace the deflectors if the clearances are excessive and oil leakage is a problem (either leaking to the outside where lagging can become soaked, creating a fire hazard, or into the steam system).

231-6.8 LEAKING OIL

231-6.8.1 PRECAUTIONS AGAINST LEAKING OIL. Any joint in the lube oil system equipment or piping is potentially a leakage source. Fortunately, most leaks develop slowly, giving ample opportunity for repair. Exceptions are ruptures of small-diameter piping (gage or instrument lines) from vibration fatigue failure. Ruptures are best avoided by liberal use of pipe hangers or by correcting the cause of vibration. Ideally, there should be no oil leakage, so stating a leakage criterion is difficult. The best rule is that if a leak becomes a housekeeping or safety problem, it must be repaired.

231-6.8.1.1 Leaks should always be repairable by use of proper gaskets, where fitted, and by proper tightening of geometrically correct mating parts. Flat flanges require sheet gaskets, and raised flanges require flexitallic gaskets.

231-6.8.1.2 The bearing bracket joint is a metal-to-metal oil-tight joint that has a thin coating of sealant. To assemble this type of joint:

1. Clean and stone the joint to remove any raised metal. Remove studs if necessary to remove raised metal near them.
2. Verify correct assembly of bearings and components so that joint will not be held open.
3. Verify correct pinch fit if a new spherically seated bearing was just installed.
4. Do a joint contact blue check if bearing or other components were changed or if the joint is known to be damaged. Blue check the joint and hand-work as necessary to achieve a minimum area of 70-percent contact inside the bolt pattern. Areas of noncontact should be distributed, not concentrated in one area only.
5. Apply a thin coat of nonhardening Permatex or equivalent sealant on the joint, stopping 1/4 inch from the edges. Excess sealant will seep into and contaminate the lube oil system. Equivalent sealants are those that have been shown to be effective in stopping leaks in these joints, and that will not form a bond to prevent joint disassembly. The silicone compound Permatex Ultrablue 77B has been shown to be effective and to allow disassembly. Other silicones, however, even in the same mil-spec or NSN, must be tested before being used. If a silicone is used, stop 3/4 inch from the inner edge and use nonhardening Permatex inside this band, stopping 1/4 inch from the inner edge.
6. Reinstall upper bearing bracket and dowels. Verify precautions in paragraph 231-6.4.6.5, as necessary. Torque fasteners in a crosswise incremental torque.

231-6.8.2 OIL LEAKAGE. There are various contributing causes for leakage and for turbines throwing oil. Most of these causes can be remedied by the operating personnel. The following instances are cited with the corrective measures to be taken:

a. Turbine oil deflectors can leak if clearances are excessive or drainage from journal-bearing brackets are blocked or restricted. Difficulties from oil-deflector leaks are infrequent if the oil system is kept clean, proper system pressure is maintained, and oil sumps are at specified levels. Careful turbine warmup and low-vibration operation will minimize deflector wear.
b. The drain holes in the bearings may be too small or become clogged with the dirt or residue in the oil. The drain holes are usually designed by the manufacturer to provide for the original bearing clearance, and as the bearings wear and the drain holes become clogged with dirt, a greater quantity of oil tends to pass through the bearings than the drain holes can accommodate. This causes the oil to back up and flow out the ends of the

bearing. The holes may be reamed out to a slightly larger diameter, and it may be advisable to widen the oil grooves slightly. The size to which these holes may be safely enlarged is limited; if the bearings are badly worn the only remedy is to fit spare bearings. The passage of the oil grooves and the drain holes should be cleaned of all sediment when the bearings are taken out for overhaul.

c. Leakage from the ends of the bearings may be caused by setting the oil relief valve too high.

231-6.8.3 DANGERS FROM LEAKING OIL. An oil leak is initially a nuisance, but it may develop into a serious problem. Small leaks can eventually become large leaks, jeopardizing equipment from lack of lube oil or permitting oil to spray on and burn from contact with hot steam surfaces. Turbine or piping lagging that becomes oil soaked can also be ignited by hot steam surfaces. At intermediate temperatures, the lagging initially smolders and emits smoke that irritates eyes and lungs. With elevated temperatures, smoldering can develop into an open flame and a full-scale fire. Pay particular attention to the low-pressure turbine during astern operation since the turbine can become quite hot. Soaked turbine lagging that remains undetected in the cool-casing condition of ahead operation can start smoldering with prolonged astern running. Short of lagging replacement, little can be done to hasten oil boiloff. A fire watch with appropriate equipment should stand by when a boiloff condition develops. Fires are doubly dangerous in submerged submarines because in addition to heat and smoke, compartment oxygen can soon be depleted.

231-6.8.3.1 Dangers from leaking oil can be avoided by properly maintaining seals and piping systems. Lagged surfaces on which oil could drip during bearing disassembly, oil strainer cleaning, or other routine maintenance actions should be protected by sheet metal covering.

231-6.9 OIL QUALITY

231-6.9.1 GENERAL. Steam turbine lubricating oil requirements conform to MIL-L-17331, Lubricating Oil Steam Turbine and Gear, Moderate Service, the military symbol for the oil is 2190 TEP. Steam turbine lubricants have a high viscosity index (VI). A VI of 90 to 100 is typical of paraffinic lubricating oils. The viscosity of an oil with a high VI (100) varies less with temperature changes than does the viscosity of an oil with a low VI (40). Paraffinic oil contains additives that inhibit rust, foam, oxidation, and sludge formation. Turbine oils shall have good water separating characteristics and shall not form stable emulsions with water, a common system contaminant. The use of additives is therefore highly selective. Lubricants with chlorinated additives are prohibited from steam turbine lubrication systems. Laboratory tests have verified that combining certain metals with lubricants containing a chlorinated additive can sometimes cause bearing failure. After-market lube oil additives (MILITEC, STP, SLICK-50, WYNNS, etc.) should not be added to lube oil systems or components. The correct additives are blended into the oil at the oil factory for the purpose intended or to meet the mil-spec or commercial spec. After-market additives have been tested and have not been shown to improve the oil and in some cases have hurt the equipment. For more information on steam turbine lubricants and their maintenance, refer to NSTM Chapter 262, Lubricating Oils, Greases, Specialty Lubricants, and Lubrication Systems.

231-6.9.2 OIL PURITY. Using clean, pure oil is essential to the long life and successful operation of a turbine. Bearings and other oil components can last the life of the ship without repairs if pure, clean oil is supplied continuously during operation. Take every possible measure to keep the oil in good condition and to prevent water, dirt, and other foreign substances from entering the system. Proper use of the lube oil purifiers is essential to oil purity.

231-6.9.3 OIL FLUSHING. As a rule, operating lube oil systems are found with clean lube oil; only after an availability or overhaul is foreign matter found. When a turbine is new, or after repairs have been made to oil components or piping, or when carbon steel piping or components have been allowed to rust, a flush is necessary. This flush should at least clean oil piping without contaminating oil components. Control components usually require disassembly and hand cleaning. Note that post-availability sea trial operations will normally jar and vibrate the lube oil system enough to break loose foreign material hidden in the system, even after flushing. To contain this foreign material, muslin or nylon bags should be kept in the lube oil strainers from initial operation through final sea trials.

231-6.10 OIL COMPONENTS

231-6.10.1 STRAINER AND FILTERS. The lube oil system normally has strainers protecting turbine bearings and finer-mesh filters protecting turbine control components. Turbine bearings can tolerate small crushable rust particles without damage. The sizing of strainers and filters should be chosen based on the component clearances being protected, the maximum flow rate needed to those components, and the strength needed to hold back the maximum possible differential pressure (DP). Strainers and filters should be inspected and cleaned when a specified DP is read across them at least annually. Take care to remove and to note any metallic particles that may adhere to the magnets in the strainer baskets. Strainers and filters are rated nominal--meaning that not many of a specified size particle can be found downstream; or as absolute--meaning that no particles of a specified size are found downstream. Note that it is prudent not to allow carbon steel pipe or components downstream of a filter since they can be a source of downstream rust particles. Filters are often sized in microns, with 30 microns approximately equal to 1 mil (0.001 inch).

231-6.10.2 BEARING LUBRICATION. Because steam turbines run at high speed, it is absolutely essential that all bearings receive the proper amount of lubrication. In addition to the pressure caused by moving parts, additional strain is thrown on the bearings when the ship rolls and pitches. On a ship rolling or pitching excessively, the additional force applied to the bearings due to gyroscopic action is enormous. The importance of using the proper oil, maintaining this oil at the proper pressure, keeping it clean (clear and bright) at all times cannot be overemphasized. Improper grooves have caused hot bubbler temperatures because not enough unworked oil is flowing to the bubbler supply.

231-6.10.3 GOVERNOR AND OVERSPEED TRIPS. All sliding contacts and pivot points in governor and overspeed tripping mechanisms shall be kept free, clean, and well oiled so that there will be no sticking to prevent the safety devices from functioning at the set speed of rotation. Pay particular attention to the condition of pins or other securing devices on overspeed tripping mechanisms.

231-6.10.4 OIL RELIEF VALVES. Relief valves are usually fitted in the lubricating system to avoid excessive pressures. The manufacturer’s special operating instructions should be followed. Relief valves at lube oil pump discharges protect the system and usually relief valves or regulating valves control the oil pressure to bearings or turbine control components.

231-6.10.5 RUSTED COMPONENTS. Lube oil system components are often made of carbon steel and are subject to rust and pitting when not in wet lay up, completely filled with lube oil. Do not allow ducts from the ship’s air conditioning system to blow on power generators and other turbine control components. These are not filled with oil and moisture from the lube oil system can condense on internal walls, cause rusting and pitting, and shorten their operating lives. See paragraph 231-9.4 for more information.

231-6.11 SHIP SERVICE TURBINE GENERATOR (SSTG) REDUCTION GEARS

231-6.11.1 GENERAL. A turbine operates economically at a high speed, but in some cases the driven generator is most efficient when operating at low speed. To derive efficient results, each unit must operate at its approximate economical speed. To effect this requirement mechanical reduction gears are used. Reduction gears used with turbine generator are helical. Helical gears are used where the shaft of the driven generator is mounted parallel to the shaft of the turbine. If the gears are alined properly and satisfactory tooth contact is obtained when the gears are first installed, little trouble should be experienced with turbine generator reduction gears, provided they are supplied with an ample quantity of clean lubricating oil. For further information on SSTG gears, refer to NSTM Chapter 241, Propulsion Reduction Gears, Couplings, Clutches, and Associated Components.

231-6.11.2 GEARS FOR SSTG'S. New gears or gears that have been realined should be thoroughly worn in at low power before being subjected to the maximum tooth pressure of full power. Burnishing or a slight wear-in pattern, particularly along the pitch line, may occur in the first few weeks of service. Burnishing usually ceases after this wear-in period. The only care necessary is to ensure that no metal flakes remain in the oil system.

231-6.11.2.1 Imbalance Indication. A properly operating gear has a certain definite sound, which the trained operator can easily recognize. Investigate any unusual noises, and operate the unit with caution until the cause is discovered and remedied. Vibration is caused by faulty alinement, bent shafts, a damaged driven generator, or improper balance. Gear wheels and pinions are balanced dynamically when the units are built. Unless damaged or corroded in service, or improperly erected, they should remain in balance. Imbalance in the gear is indicated by unusual vibration, noise, or unusual bearing wear. The vibration may, however, be due to imbalance in the turbine or in the driven generator. The turbine rotor is much more likely to be out of balance than the gears.

231-6.11.2.2 Bearing Wear. Use the bearing section of NSTM Chapter 241 for guidance when inspecting, renewing, or fitting SSTG gear bearings. The amount of gear bearing wear shall not be allowed to become sufficiently great to cause incorrect tooth contact. (See paragraph 231-8.14 for rebabbitting of bearings.) It is essential, for proper operation of the gears, that the total tooth pressure be uniformly distributed over the total length of the tooth faces by accurate alinement and adherence to the designed clearances. The designed center-to-center distance of the axes of the rotating elements should be maintained as close as practical, and in all helical gears the axes of the pinion and gear shafts shall be parallel. Nonparallel shafts concentrate the load on one end of a helix and may cause a feather edge on the teeth or flaking, galling, pitting, or deformation of tooth contour; or may break the ends of the gear teeth.

231-6.11.2.3 Oil Supply. All gears are fitted with spray nozzles that spray oil on the meshing teeth. These spray nozzles shall be kept open at all times.

231-6.11.2.3.1 Proper initial alinement, an adequate supply of lubricating oil at all times, proper clearances, and inspection for damages should prevent trouble from wearing and scoring of teeth. If, after all precautions, the lubricating oil supply should fail and the teeth become scored, the gears shall be overhauled at the first opportunity.

231-6.11.2.4 Gear Teeth. Use the tooth contact section of NSTM Chapter 241 for guidance when checking tooth contact of SSTG gears. The teeth should never be dressed as long as the gear is operating satisfactorily, not even in cases of excessive abrasion, until after every attempt has been made to rectify the alinement. The abraded portions of the teeth should be dressed only sufficiently to prevent meshing teeth from scoring. Use steel scrapers or a fine file, and take every precaution to remove all emery, filings, or abrasive material. Make no attempt to remove deep pitting or galling. In the case of helical gears where both pinion and gear teeth have been damaged by foreign bodies, both should be relieved of all humps. Further spotting and scraping should be contained to the teeth of the gear wheel. The high spots will show more quickly on the pinion because the pinion comes in contact more often. The instruction manual for a particular installation will usually prescribe the method and equipment for determining the alinement of gears. In service the best indication of proper alinement is good tooth contact and quiet operation.

231-6.11.2.4.1 Tooth contact can best be checked by treating the pinion teeth with a light coating of metal marking, preferably Prussian blue, red lead, or copper sulphate, and rotating the pinion in mesh with the gear. Observe the tooth contact and adjust if necessary. For helical gears 80 percent of the face width should make contact. The contact should be uniformly distributed along the entire face width of the tooth. Necessary adjustments may include rebabbitting or renewing the bearings or scraping the bearings. Avoid general scraping of teeth, as this is likely to make matters worse instead of better. Do not hesitate to obtain technical assistance.

231-6.11.2.5 Inspections. Gear cases are sometimes provided with inspection plates so that rotating parts can be sighted. Examine tooth contact and condition at regular intervals at the discretion of the Engineering Officer. Before removing the inspection covers, the vicinity of the opening must be free of all nuts, bolts, tools, or other foreign matter that might fall into the gear case and cause damage. While gear cases are open the openings shall never be left unattended unless satisfactory temporary closures have been installed. Before replacing an inspection cover, connection, or fitting that permits access to the gear casing, the responsible officer shall carefully inspect the casing to ensure that no foreign matter has entered or remains in the gear casing.

231-6.11.2.6 Lay-Up. When an SSTG with gears will be secured for a period of time, use the securing and lay-up procedures of NSTM Chapter 241 for guidance in maintaining the gears.

231-6.12 TURBINE INTERNALS

231-6.12.1 INSPECTING. Only a limited number of internal turbine parts can be inspected without lifting the turbine cover or breaking high-pressure joints. Access through bolted-on manhole or handhole openings is normally provided in the exhaust end of the high-pressure turbine and the inlet and exhaust ends of low-pressure turbines. The occasions for removing turbine inspection covers would include inspecting internals before overhaul and at times when operating conditions suggest internal damage exists. Manhole or handhole openings permit sighting of the last row of rotating blading for high-pressure, low-pressure, and astern turbines. Inspect the blading for cracks; integrity of stellite shields, if fitted; corrosion pitting; erosion; lifting of blade shrouding; or evidence of foreign material having gone through the blading. Structural elements, such as struts, tie plates, and ribs, should be inspected for cracks, particularly at weld joints.

231-6.12.2 DEPOSITS. Powdery deposits that can easily be cleaned off blades or internal surfaces by hand are normally boiler carryover products. In the main they are water soluble and will be self-cleaning in parts of the turbine operating in the steam moisture region. Deposits can be a problem in superheated sections of the turbine in propulsion plants lacking superheat control. The immediate effects of deposits are to restrict steam passages and cause higher than normal pressure and reduced efficiency. The long-term effect, if deposits contain salt (sodium chloride), is corrosion with pitting of blade vane sections.

231-6.13 APPROVAL FOR LIFTING TURBINE CASING

231-6.13.1 REASONS FOR LIFTING. Where steam and oil are kept clean and proper drying procedures are followed (Section 9), deterioration of blading and other internal parts is practically eliminated. Many steam turbines last their 30-year lifetime without major overhaul. Lifting a turbine cover will cost hundreds of thousands of dollars, and internals can actually be damaged during overhaul. A request for lifting a turbine cover, therefore, shall be for cause other than accumulated time or operating hours. A request to lift casings should be made only when knowledge or suspicion of internal damage or hazard justifies lifting. All other means short of lifting, such as inspection and diagnosis, should be used before making a request for lifting. Casing vertical steam joints are designed never to be disassembled. If necessary, however, OEM technical assistance is required.

231-6.13.2 TECHNICAL DETERMINATION. To minimize the number of unnecessary casing disassemblies, technical determination of the necessity for casing disassembly should be obtained directly from NAVSEA. All correspondence regarding approval for casing disassembly shall be forwarded to NAVSEA for concurrence or approval. An existing ship alteration (SHIPALT) or other NAVSEA correspondence requiring casing lifting may be taken as approval.

231-6.13.3 FINAL AUTHORIZATION. The Type Commander shall provide the final authorization for lifting casings on the basis of NAVSEA technical evaluation of conditions reported, schedules, and funds available. Conditions found when opening a turbine casing shall be documented in the turbine opening and closing reports along with suitable photographs or video tape of the regions inspected.

231-6.14 EXAMINATION OF FAILED PARTS

231-6.14.1 DETERMINING THE CAUSE OF FAILURE. Whenever parts are found damaged or failed and the cause is not improper maintenance, faulty operation, or normal wear, the failure should be considered a result of either an inappropriate design or faulty manufacturing. The failed part shall be thoroughly examined and, when necessary, preserved for further examination to determine the cause of failure. Notify NAVSEA of the failure, giving circumstances and detailed results of examinations. NAVSEA may desire further investigation of the failed part by Naval Activities or by the original manufacturer; however, parts need not be held for more than 2 months after reporting the failure.

231-6.14.2 PHOTOGRAPHS OF FAILED PARTS. Take photographs of the damaged parts to supplement verbal descriptions. Photographs are particularly helpful in interpreting hard-to-describe or unusual markings, such as those associated with corrosion or erosion damage. The photograph becomes a permanent record that can be compared with photographs previously taken and used to establish a rate of degradation, where appropriate. This is often of primary importance in establishing the significance of the condition.

231-6.14.2.1 Identifying Parts in Photographs. Although there is no doubt as to what is being photographed at the time of taking pictures, it can become indistinct later and could be completely unrecognizable to persons reviewing reports. Including the following items in the picture will help orient the viewer:

1. Place a scale, preferably a 12-inch ruler, next to the part.
2. Place a placard with the part name next to the part, or mark the part itself, circling the damaged area with contrasting color. Nomenclature should correspond to that of the technical manual (for example, 5th Stage Diaphragm, Upper Half, Steam Exit Side, High-Pressure Turbine, Port Unit, DDG 40, 26 August 1986).

231-6.14.2.2 Photographs of Fine Details. Use professional photographers with closeup cameras if fine details are important. Early stages of erosion or corrosion damage are in this category, as are fracture surfaces where fatigue can be associated with typical marks. Taking several photographs at different distances and positions will permit later selection of the best picture.

231-6.15 PRECAUTIONS AGAINST ENTRY OF FOREIGN MATERIAL

231-6.15.1 GENERAL. Foreign material should not be allowed to remain in any steam turbine internals or turbine steam component internals, since this could potentially contaminate the boiler or steam generator. For nuclear ships, refer to NAVSEAINST 9210.36, Steam Plant Cleanliness Control, for availabilities or NAVSEA 0989-064-3000, Cleanliness Requirements for Nuclear Propulsion Plant Maintenance by Forces Afloat, for maintenance by Forces Afloat. This section supplements these documents that give general guidelines for personnel entering turbine exhaust areas for inspections. For Foreign Material Exclusion (FME) from lube oil components, see paragraph 231-6.9.

231-6.15.2 APPROVAL TO ENTER TURBINE. For approval to lift the turbine casing, see paragraph 231-6.13. No opening giving access to the inside of a turbine casing shall be uncovered without prior approval of the ship's Engineering Officer. Openings shall not be left unattended unless suitable temporary covers are installed. Surveillance shall be maintained at all times to prevent unauthorized tampering with covered or uncovered openings. Ensure that ventilation of the turbine interior is adequate and a second person is at the opening before allowing personnel to enter.

231-6.15.3 CLEAN AREA. Set up a clean area before lifting any turbine casing, or opening any cover giving access to the inside of the turbine casing or to a steam component, bearing, or lube oil component. Clean the area above and around the cover to be opened. Tape up appropriate plastic material as a barrier to prevent foreign material from falling in. An enclosed tent is not necessary unless required by the ship's Engineering Officer.

231-6.15.4 TOOLS AND PARTS. Strict accountability for tools and parts shall be maintained when working in turbines to ensure that nothing foreign is left inside. Personnel who enter steam turbines or work and inspect around uncovered openings should not carry objects on their person that can come undone and fall into equipment. Secure flashlights, tools, and eyeglasses with lanyards. Empty pockets; remove badges, belts, shirts with buttons, and any other object that could fall into the turbine or component. Taping shoes, zippers, and pants rivets is unnecessary. Shoes shall be wiped as clean as possible, or clean cloth covers put over shoes before entering the steam turbine exhaust area.

231-6.15.5 TURBINE EXHAUST INSPECTION. Personnel entering a turbine exhaust area for any inspection should take a little additional time to do a general inspection of the area for obvious problems. If in doubt about an observation, obtain technical assistance from local personnel or see paragraph 231-6.13. Other documents that apply include NSTM Chapter 254, Condensers, Heat Exchangers, and Air Ejectors and 9254-SMMS-GYD, Main Condenser Inspector's Guide. Things to inspect include:

a. Exhaust boot interior surface for obvious tears, delamination, wear, or oil residue.

b. Turbine casing drain pipes, with unions or brackets, are intact and are still connected, and fasteners are not hand-tight.

c. Spray pipes show no cracks when pulled on. Nozzles are attached and appear unobstructed.

d. Last-stage turbine blades and shrouds are not missing or cracked. Most steam turbines have at least one locking piece in the blade row, which looks like a blade skirt without the blade. Every shroud band has a space between itself and the next shroud band. Blades by GE have silver plating partway up the blade.

e. Accessible internal fasteners are not loose or missing and their locking devices are intact and engaged.

f. Astern turbine internal casing drain orifice fitting is intact.

g. There are no obvious visual cracks in structural welds.

h. There are no obvious carryover deposits or foreign material.

231-6.15.6 PRECLOSING INSPECTION. Immediately before permanently closing an opening that has been uncovered, the ship's Engineering Officer or his representative shall inspect the turbine through the opening and inspect any records to verify that neither foreign material nor unsatisfactory conditions exist. Record this inspection and the name of the inspector in the engineering log.

231-6.15.7 REPORTING FOREIGN MATERIAL. If, despite all efforts, an object is accidentally dropped into a turbine, immediately report this to the Engineering Officer. Make all reasonable efforts to retrieve the foreign material.

231-6.15.8 RETRIEVING FOREIGN MATERIAL. Lifting the upper turbine casing is the ultimate and sure method of recovering foreign material. Permission should be requested per paragraph 231-6.13 when all other attempts fail. These attempts should include:

a. Reaching an object through inspection covers.

b. Removing crossover pipe valving or other piping.

c. Using a magnet, forceps, hooks, or adhesive on the end of a stiff or flexible extension rod or boroscope.

d. Blowing out the object using air pressure or sucking the part from the turbine using a vacuum cleaner and small tubing as necessary.

231-6.15.9 INTERNAL COATINGS. No coatings, paints, or varnishes are permitted on internal steam or oil surfaces of an operating steam turbine. Remove preservatives of MIL-T-17286, Turbines and Gears, Shipboard Propulsion and Auxiliary Steam; Packaging of, from lay-up periods before turbine operation or during initial flushes. See paragraph 231-6.9 for information on short-term lay-ups and Section 9 for information on long-term lay-ups.

231-6.16 INSTRUMENTATION

231-6.16.1 GENERAL. Main gage board instrumentation gives readouts on system pressures and temperatures requiring more or less monitoring when operating main turbines. At this central location the status of the main steam and various support systems can be determined by comparing pressures and temperatures with established normal values for various modes of operation. Other direct-mounted instrumentation is frequently provided to give more detailed information on a particular piece of equipment, rather than on the system as a whole. Suggested intervals for reading instruments are given in paragraph 231-7.1.1.

231-6.16.2 PERIODIC CALIBRATION. Establishing normalcy by instrumentation usually requires only that the gages be consistent; that is, the gage should have similar readings with the same input at different times. If the equipment or systems are to be checked against design values or if gages in a system are to be checked relative to each other, the gages must be in calibration. For calibration the gages are removed and checked against a standard, such as a deadweight gage or another gage known to be in calibration. Gage reading, adjusting for waterleg, converting gage pressures to absolute pressures, calibration type, and frequency of gages normally used in turbine systems are discussed in detail in NSTM Chapter 504, Pressure, Temperature, and Other Mechanical and Electromechanical Measuring Instruments.

231-6.16.3 PRECAUTIONS AGAINST FALSE INDICATION. In most cases incorrect readings from gages that are grossly incorrect are easily spotted by obvious inconsistencies with previous experience at the existing condition. Gages with marginal inaccuracies are more difficult to spot and may lead to the wrong conclusion as to equipment fitness. Before any conclusion is reached, establish that the gage and gage line are satisfactory.

231-6.17 REPAIR PARTS ADMINISTRATION

231-6.17.1 TECHNICAL MANUAL LISTS. Newer turbine manuals contain a list of Onboard Repair Parts and Tools; older manuals may also contain a list of Shore-Based Repair Parts. The lists show repair parts that were furnished or made available to the ship under the original contract or purchase order for the turbine. The lists are not the authority for carrying specific onboard repair parts, but they may be useful for locating the repair part nomenclature and part number. Normally, a Navy Drawing List or Turbine Drawing List is included in the manual. The tabulation lists turbine part nomenclature, alphabetically and arranged cross-referenced to plan and piece number. This is useful where plan and piece numbers are required to identify a specific repair part.

231-6.17.1.1 Technical manuals provided to auxiliary or amphibious ships, converted or built to basic Maritime Commission designs, usually do not contain repair parts lists. In these cases onboard drawings (if available) and the ship allowance lists are the best sources for identifying the repair part or special tool.

231-6.17.2 ALLOWANCES. Repair parts required for maintenance are identified in either the Coordinated Shipboard Allowance List (COSAL) or the Ship Non-Tactical ADP System (SNAP). COSAL is a hard copy document; SNAP is an automated system.

231-6.17.2.1 Both COSAL and SNAP include all equipment on board the ship, including steam turbines, and indicate:

a. Authorized repair parts, special tools, clothing, and other materials required on board to support the installed equipment

b. Technical and descriptive data of the installed equipment.

231-6.17.2.2 COSAL and SNAP are particularly useful because they contains Allowance Parts Lists (APL). Each APL identifies all the repair parts for a specific piece of equipment, whether it is stocked on board or not, in part number or manufacturer’s code number sequence. The National Stock Number (NSN) for each repair part is also listed in the APL.

231-6.17.2.3 A frequently needed repair part or tool, unlisted on the authorized onboard allowance list, may be recommended for an allowance by submitting an Allowance Change Request, NAVSUP Form 1220-2. This is submitted through the Type Commander in accordance with NAVSEAINST 4441.2, Changes to Coordinated

Shipboard Allowance Lists (COSALS); Procedures For. Complete instructions on recommending an item to be added or deleted as an On Board Repair Part (OBRP) may be found in SPCCINST 4441.170, COSAL Use and Maintenance Manual.

231-6.17.2.4 Diaphragms and nozzle blocks for propulsion or SSTG steam turbines are extremely rugged and last for the life of the ship. Even if needed for a turbine casualty, procurement lead times are relatively short. Per NAVSEA letter serial 56X21/462 dated 14 November 1984, these parts should not be procured as spare parts.

231-6.17.3 ORDERING. All required repair parts should be ordered from the appropriate stock point in accordance with Military Standard Requisitioning and Issue Procedures (MILSTRIP) outlined in NAVSUP 437, (ashore activities) and NAVSUP 435 (afloat activities), Milstrip/Milstrap Operating Procedures Manual. Each differently identified part shall be ordered on a separate requisition (form DD1348) using the appropriate 11-digit ordering number. The number can be obtained from the applicable COSAL or the catalogs of the appropriate stocking point.

231-6.17.4 REPAIRABLE PARTS. Steam turbine rotors and journal bearings are classified as depot level repairable parts. When these parts are replaced in a turbine, the failed part is returned to the stock system for a credit and is overhauled to new condition at a designated overhaul point (DOP). Repairing a repairable part is usually much less expensive than manufacturing a new one.

231-6.17.5 DESIGNATED OVERHAUL POINT. The DOP is authorized by NAVSEA to overhaul a depot level repairable part. The DOP is normally the OEM, and this is the case for steam turbine rotors. For journal bearings the DOP can be the OEM of the bearing or any other qualified bearing manufacturer who has access to the necessary bearing drawings. Any DOP should have the following items:

1. Overhaul facilities necessary for the rotor or other part. Class B overhaul (defined in paragraph 231-8.1.1.2)
2. The engineering support necessary to evaluate conditions and corrective actions, including access to drawings and procedures
3. A quality assurance plan, as necessary
4. Necessary government inspection services

231-6.17.6 ROTOR INSPECTION AND REPAIR GUIDELINES - STEAM TURBINE. Rotors sent to the DOP for overhaul to Class B condition should meet the inspection and repair guidelines given in Table 231-6-4 and Table 231-6-5. Turbine rotors are critical components as defined in paragraphs 231-6.17.5 and 231-6.17.8. Table 231-6-4 and Table 231-6-5 do not change the requirement to repair rotors at the DOP by the OEM. Repair guidelines such as 4b, 4c, 4d, 6b5, and 6d1 in Table 231-6-5 must have OEM engineering analysis, which can only be found at the OEM factory. Waivers from the specific requirements in these tables should be handled through normal channels on a case-by-case basis.

231-6.17.7 DEFECTIVE PARTS. When a defective part is received from the stock system, the repair activity should fill out and submit a Quality Deficiency Report (QDR), SF364 form, in accordance with the directions on the form. This will be forwarded to the Naval Inventory Control Point (NAVICP) (formerly Ships Parts Control Center, SPCC, Mechanicsburg, PA) Fleet Material Supply Office, where a case number is assigned. The case number will be tracked until resolution. Always retain the part shipping papers in case they are needed for the QDR. When the QDR is resolved, NAVICP will forward the answer to the repair activity.

231-6.17.8 CRITICAL SPARE PARTS. A critical spare part is defined by Chapter 141 of title 10, United States Code, section 2383 as any spare part that is critical to the operation of an aircraft or ship. Also stated is that a critical spare part shall meet the same quality and inspection requirements as the original part. Steam propulsion and power generation are required for the operation of a steam-powered ship. On the basis of this, critical parts for propulsion and SSTG steam turbines are defined as rotating parts (such as rotors, blades, emergency governor, or overspeed trip parts) or control parts (such as control valves, governor valves, trip throttle valves, and parts for these valves and governors). Any defective turbine critical spare part shall be reported as described in paragraph 231-6.17.7. Send a copy of the QDR to NAVSEA.

231-6.17.8.1 The critical parts defined in paragraph 231-6.17.8 are also critical to personnel safety since they are designed to prevent uncontrolled overspeed and destruction of the steam turbine.

231-6.17.9 OVERSIZED PARTS. Parts such as valve seats, bushings, and fitted bolts are often stocked as oversized parts. Mating surfaces can be remachined, and the oversized part machined to fit before installation. Check COSAL's and OEM drawing lists to confirm if oversized parts are stocked.

231-6.17.10 STORAGE. Store steam turbines, SSTG's, main propulsion complexes, or any parts or assemblies or components of these indoors in a dehumidified building. Both oil and steam side internals shall be preserved with the proper preservative in accordance with contract instructions. Lift rods should be properly preserved in the area that remains in the lift rod bushings to prevent corrosion and seizing. When transporting outdoors or sitting outdoors awaiting installation, they shall be covered by a waterproof tarp and set off the ground. Handle this equipment with care to avoid damage.

231-6.17.11 IN-STORAGE MAINTENANCE. Periodically inspect equipment that is in storage. Verify that dehumidifiers are still working, that components left with oilside wet lay-up have not leaked, and that corrosion has not started. Add the proper preservative to the valve lift rod bushing clearances, and exercise the lift rods (or valve stems) to work in the preservative. Main propulsion complexes and geared SSTG's require weekly rotation of rotors (**NSTM Chapter 241**).

Table 231-6-4 STEAM TURBINE ROTOR INSPECTION GUIDELINES

Inspection or Action Item	Acceptance Criteria or Corrective Action
1. Identify rotor.	Inspect the rotor that was received to ensure that it is the same as the one identified in the contract.
2. Record as-received condition.	a. Inspect for missing or damaged parts. b. Inspect for unusual deposits and retain a sample. c. Inspect for balance weights. d. Take photographs.
3. Clean rotor.	Mask and blast or clean without damage to aid in inspection. Record the method used.
4. Perform nondestructive testing (NDT).	NDT per MIL-STD-271 requirements. a. Magnetic Particle Test (MT) the entire rotor. b. Do an MT of each blade. c. Do an MT of each shroud and tenon. d. Do a Liquid Penetrant Test (PT) if required by the drawing. e. Do an Ultrasonic Test (UT) if required by the drawing
5. Check runout.	a. Verify that the rotor is not bowed and record the results. b. Verify that journals are round and concentric; record verification. c. Verify and record thrust collar runout.

Table 231-6-4 STEAM TURBINE ROTOR INSPECTION GUIDELINES - Continued

Inspection or Action Item	Acceptance Criteria or Corrective Action
6. Measure critical dimensions.	a. Measure journal diameters and taper. b. Inspect journal finish. c. Inspect thrust collar finish. d. Measure coupling surfaces, dimensions, and fit. e. Inspect packing or oil deflector flat lands. f. Inspect packing high-low lands.
7. Check rotor balance.	a. Low-speed balance is allowed. b. Refer to Repair Guidelines, Item 7b in Table 231-6-5.
8. Visually inspect and record results.	a. Inspect the whole rotor for unusual damage. b. Inspect blades for nicks, rubs, erosion, corrosion,tenons raised from shroud, missing or cracked blades, and missing or cracked tenons. c. Inspect shrouds for nicks, rubs, erosion, corrosion, or shroud that is missing, cracked, or lifted from the blade tip. d. Inspect wheels for watercut or erosion.
9. Trip assembly.	Inspect to drawing without disassembly. Verify that moving parts move freely.
10. Pack and ship.	Refer to Repair Guidelines, Item 10 in Table 231-6-5.

Table 231-6-5 STEAM TURBINE ROTOR REPAIR GUIDELINES

Inspection or Action Item	Acceptance Criteria or Corrective Action
1. Identify rotor.	Refer to Inspection Guidelines, Item 1 in Table 231-6-4.
2. Record as-received condition.	Refer to Inspection Guidelines, Item 2 in Table 231-6-4
3. Clean rotor.	Refer to Inspection Guidelines, Item 3 in Table 231-6-4.
4. Perform NDT.	a. If a major MT problem is found on the rotor, repair it with a NAVSEA-approved procedure, or recommend scrapping. If a minor MT problem is found, (NAVSEA 0900-LP-003-8000) grind it out.
	b. Chase minor MT findings per OEM engineering analysis; replace blade or blade row for major MT findings.
	c. Chase minor MT finding per OEM engineering analysis; replace blade group or blade row for major MT finding.
	d. Repair per OEM engineering analysis.
5. Check runout.	a. Bowed rotor - Repair with a NAVSEA-approved procedure, or recommend scrapping.
	b. Reestablish drawing journal roundness and concentricity within drawing journal diameters and tolerances. If this cannot be done, do not undersize journals, but chrome-plate per DOD-STD-2182 and machine back to drawing dimensions.
	c. Reestablish thrust collar runout by machining within design dimensions and tolerances. Collar thickness must allow successful thrust bearing shimming.
6. Measurement critical dimensions.	a. Reestablish drawing journal diameters and lack of taper as described in item 5b above.

Table 231-6-5 STEAM TURBINE ROTOR REPAIR GUIDELINES -

Continued

Inspection or Action Item	Acceptance Criteria or Corrective Action
	b. 1. Reestablish drawing journal surface finish only if it is specifically required by the contract. 2. Surface discolorations that have no depth require no action, except that bluing (from heat) requires OEM identification and recommendation to NAVSEA. 3. Circumferential running marks (1 mil or less in depth) appearing to be caused by fine dirt in the oil, and that would not cause babbitt wear, require no action. 4. Small pits or dings (less that 2 mils in depth) or slight random scratches (less than or equal to 1 mil in depth) require no action. 5. Remove any raised metal by using an OEM-approved procedure without losing minimum journal diameter. 6. Deep phonographing, scoring, or grooving of journals (greater than 1 mil) that might cause babbitt wear and easily catches a fingernail requires journal repair. Machine journal, chrome-plate to DOD-STD-2182, and machine back to drawing dimensions. 7. Journal damage exceeding 0.025 inch radially requires repair by machining the journal to a standard undersized diameter (corresponding to pre-existing special bearings). If no pre-existing special bearings exist, machine the journal to 0.100-inch undersized diameter as a standard. Pack a special bearing with the rotor. Document the undersized journal on the configuration drawing and send deviation drawings to NAVICP and NAVSEA. 8. More severe journal damage requires NAVSEA approved repair procedures or recommendation for scrapping.
	c. 1.Reestablish drawing thrust collar surface finish by machining within design limits and parameters so that shims may be used at installation. 2.Thrust collar damage exceeding the design limit (for shimming) thrust collar thickness requires replacement or recommendation for scrapping.
	d. 1. Reestablish drawing coupling surfaces dimensions and fit by machining, and OEM-approved rolling procedures, where possible. 2. Coupling surfaces that cannot be restored by machining require OEM-approved repair procedures.
	e. 1. Remove any raised metal without losing minimum land diameter. 2. Deep grooving or valleys (greater than 1 mil in depth), or severe erosion or corrosion damage that would degrade performance of the packing lands or oil deflector flat lands require repair. Machine flat land, plate to DOD-STD-2182 with a plating metal compatible with the packing or deflector material, and machine back to drawing dimensions. 3. For lands where plating per 6.e.2 is not possible, machine the land to a standard presently undersized diameter (or 0.100-inch undersized diameter), and provide deviated packing or seals packed with the rotor. Document this on the deviation drawing with copies to NAVICP and NAVSEA.

Table 231-6-5 STEAM TURBINE ROTOR REPAIR GUIDELINES - Continued

Inspection or Action Item	Acceptance Criteria or Corrective Action
	f. 1. Remove any raised metal without losing minimum high-low land dimensions. 2. Any grooving or server erosion or corrosion damage that would degrade performance of the packing high-low lands requires machining to standard undersized high-low diameter (corresponding to pre-existing special packing). If no pre-existing special packing exists, machine the high-low land to an undersized high-low diameter and design, per OEM engineering analysis, and provide special packing with the finished rotor. Document this on the deviation drawing with copies to NAVICP and NAVSEA.
7. Check rotor balance.	a. Low-speed rotor balance is allowed when appropriate per OEM engineering analysis. Balance weights shall not be changed on a rotor certified to be high-speed balanced.
	b. High-speed rotor balance is required whenever substantial material is added to or removed from the rotor. This includes any removal or replacement of blades, shrouds, balance weights, thrust collar, stub shaft, or similar rotor components. This also includes machining or building up of journals, lands, wheels, or other rotor components. This does not include minor dressing or removal of high metal if excluded by local engineering analysis. Contract out high-speed balance if not available locally. Waivers from this requirement or clarification for a specific rotor condition may be addressed to NAVSEA on a case-by-case basis.
8. Visually inspect and record results.	a. Photograph unusual damaged areas.
	b. 1. Dress blade surfaces per OEM engineering analysis to remove nicks, dents, rubs, or minor erosion and corrosion. 2. Replace the blade row if any blade is missing, torn, cracked, or has a stress riser identified by OEM engineering analysis; is severely eroded or corroded (more than 10 percent of blade cross section is missing); or if any blade has missing or cracked tenons or tenons with pitting or MT or PT indications that would compromise that tenon’s strength to hold the shroud. 3. Tenons raised from the shroud may be repeened per OEM engineering analysis.
	c. 1. Dress shroud surfaces per local engineering analysis to remove nicks, dents, rubs, or minor erosion and corrosion. 2. Replace the blade row if any shroud is missing, torn, cracked, lifted from the blade tip, or has shroud erosion and corrosion greater than 10 percent of cross-section; or there are pitting or MT or PT indications near tenons that would compromise the shroud’s strength in that area.
	d. Watercut or erosion of rotor wheels shall be left as found or repaired per OEM engineering analysis, or scrapping may be recommended.
9. Overspeed trip assembly.	If the rotor has an integral centrifugal overspeed trip assembly, verify drawing dimensions and parameters as possible without disassembly.
10. Preserve, pack, and ship.	Per MIL-T-17286 or specific contract.

SECTION 7

TESTS, INSPECTIONS, RECORDS, AND REPORTS

231-7.1 GENERAL

231-7.1.1 PURPOSE. This section describes tests, inspections, records, and reports needed for steam turbines. Turbines receive scheduled preventive maintenance in accordance with the Planned Maintenance System (PMS) in effect fleetwide. The PMS designates specific action to be taken to ensure proper equipment operation and extended life. PMS instructions have been generated from the applicable equipment technical manuals; however, any conflicts should be reported through feedback reports for final resolution by the Carderock Division Naval Surface Warfare Center (CDNSWC), Philadelphia. Recommended periodic tests and the frequency of tests and inspections are listed in Table 231-7-1. When applicable, PMS Maintenance Requirement Cards (MRC) supersede the information in Table 231-7-1. For testing control valves, refer to paragraph 231-8.2.3.

Table 231-7-1. PERIODIC TESTS AND INSPECTIONS

Recommended Tests and Inspections	Frequency
Operating and lubricate all valve operating linkage.	Monthly
Take depth micrometer readings on the turbine journal bearings.	Quarterly
Lift sentinel and relief valves by hand.	Quarterly
Measure thrust clearance of turbines.	Annually
Inspect interior of closed turbine casing.	Annually
Clean, inspect, and preserve exterior of turbine casing.	Regular overhaul cycle
Inspect turbine bearings, journals and oil deflectors; measure clearances.	Regular overhaul cycle
Inspect shaft packing and journal for condition; check clearances.	Regular overhaul cycle
Check foundation bolt tightness.	Regular overhaul cycle
Drain and refill operating control gear boxes.	Regular overhaul cycle
Remove and clean main steam strainer.	Regular overhaul cycle
Measure nozzle clearance.	Regular overhaul cycle
Test turbine sentinel valves.	Regular overhaul cycle
Test speed limiters.	Annually
Test overspeed trips.	Annually
Test steam valve leakage.	Semiannually
Test turbines at dockside.	Regular overhaul cycle
Test turbines during sea trials.	Regular overhaul cycle

231-7.2 BEARING LOGS.

231-7.2.1 GENERAL. Use a copy of Figure 231-7-1 or a similar form to record journal bearing depth micrometer readings taken during periodic PMS checks or to record measurements taken when a bearing is opened for inspection or replaced. This log will help in troubleshooting bearing problems and show trends in bearing wear, helping the ship to anticipate repairs or replacement. The log can be used by all ships with propulsion, ship service turbine generator (SSTG), or coolant turbine generator steam turbines. It can be adapted and used for steam turbine thrust bearings.

231-7.2.2 FILLING OUT THE LOG. To fill out the log (Figure 231-7-1), complete the following entries:

a. Name of ship and hull designation and number

b. Bearing location such as main turbine #1 forward journal bearing (MT#1 Fwd)

c. Installed clearance (IC) is the bearing clearance when this bearing was installed. It can be found in original equipment manufacturer (OEM) clearance drawings or in a report from a repair facility that replaced this bearing.

d. Bearing replacement clearance (RC) is the clearance when this bearing needs to be replaced. It is taken from the turbine technical manual or the bearing detail drawing.

e. Depth constant (DC) or bearing constant is the depth micrometer reading taken when this bearing was installed, alined, and bolted down. It is found stamped on the bearing bracket near the depth micrometer hole.

f. Depth micrometer reading (MR) is taken periodically with a depth micrometer to help calculate bearing wear. See Figure 231-7-2.

g. Wear (W), bearing wear, is calculated as W= MR-DC.

h. Bearing clearance (BC) is the actual diametral clearance between the bearing and the journal at rest and is calculated as BC = IC+W. If BC is equal to or greater than RC, replace the bearing. Note that for some bearings, the maximum allowed wear (Wmax) is given directly by the manufacturer. In that case, record Wmax in place of RC, and when W is equal to or greater than Wmax, replace the bearing.

i. Name and date - the data taker and the date data was recorded.

j. Comments - On such things as the serial number (if any) of this bearing, date of any bearing replacement, nonstandard dimensions, unusual conditions of journal or bearing (if opened), any additional readings taken, recent temperatures for this bearing, or any other notes that may be helpful in the future.

231-7.2.3 Depth Micrometer Readings. Depth micrometer readings indicate journal bearing wear. Most bearing housings or brackets have a machined flat surface at the top around the depth micrometer hole. A plug is installed, protecting the hole from contamination. It also protects the machined surface from nicks and raised metal so that depth micrometer readings are accurate. A depth constant (or depth gage constant or bearing constant) is stamped on the bearing bracket or on a plate attached to the bearing bracket. This depth constant gives the depth micrometer reading when the bearing was new. Take a depth micrometer reading as follows:

1. Verify that the turbine is secured and the lube oil system is secured and cooled to ambient temperature.

2. Clean the area around the depth micrometer hole plug and remove it, inspecting the machined surface for cleanliness and nicks or raised metal.

3. Insert the depth micrometer into the hole with its flange flat on the machine surface. See Figure 231-7-2.

4. Screw the micrometer shaft down until it contacts the top of the shaft and take the micrometer reading.

5. Remove the micrometer and turn back the micrometer shaft.

6. Repeat steps 4 and 5 until two readings are the same.

7. Record the micrometer reading in the bearing log and use it to calculate the bearing wear and the bearing clearance; compare it with the bearing replacement clearance to make decisions about replacement.

BEARING LOG

a USS ______________________ HULL ______________

b BEARING LOCATION ______________________________

BEARING DESIGN DATA

c INSTALLED BEARING CLEARANCE (IC) __________________

d BEARING REPLACEMENT CLEARANCE (RC) ________________

MEASUREMENT RECORD DATA

e. DEPTH CONSTANT (DC)	f. DEPTH MICROMETER READING (MR)	g. WEAR W = MR–DC	h. BEARING CLEARANCE BC = IC+W	i. NAME AND DATE
COMMENTS:				

e. DEPTH CONSTANT (DC)	f. DEPTH MICROMETER READING (MR)	g. WEAR W = MR–DC	h. BEARING CLEARANCE BC = IC+W	i. NAME AND DATE
COMMENTS:				

e. DEPTH CONSTANT (DC)	f. DEPTH MICROMETER READING (MR)	g. WEAR W = MR–DC	h. BEARING CLEARANCE BC = IC+W	i. NAME AND DATE
COMMENTS:				

Figure 231-7-1 Bearing Log

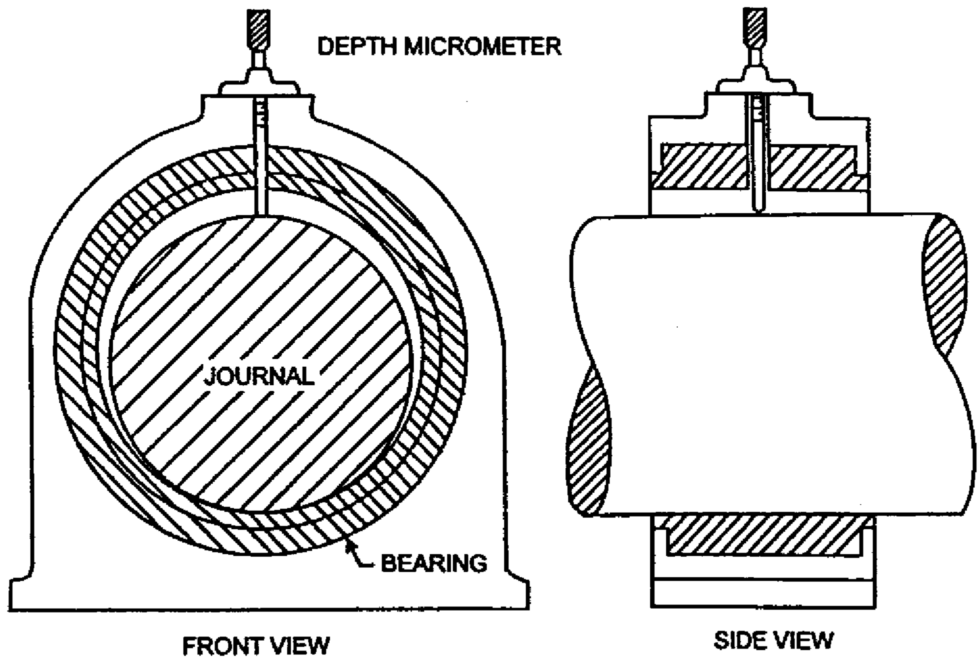


Figure 231-7-2 Depth Micrometer Reading

231-7.3 REPORT BEFORE REGULAR OVERHAUL

231-7.3.1 General. The extent of turbine work to be done in overhaul depends to a considerable degree on past operating experience and the condition that can be surmised from various measurements taken by the ship's force.

231-7.3.2 TURBINE HISTORY. The propulsion and SSTG turbine history shall contain a complete record of the turbine installation from the time of commissioning. This will help in developing the overhaul work package. The building yard should provide the first turbine history documents for each unit, with complete manufacturer's data on the unit, including size, type, and date of installation. This should also include the original rotor clearances, bearing clearances, and thrust settings and clearances. For overhaul or repair the original data must be at hand. An accurate record of all repairs, adjustments, readings, and casualties shall be kept in the turbine history. Necessary details are discussed in paragraphs 231-7.3.3 through 231-7.3.5.

231-7.3.3 WHEN TO SUBMIT PREOVERHAUL REPORT. Three months before each overhaul, submit a report on the preoverhaul condition of the propulsion turbine or SSTG to the Type Commander with information copies to the Naval Sea Systems Command (NAVSEA) and CDNSWC. In addition, forward a copy of the report to the overhaul activity and to the Naval Sea Support Center, Pacific (NAVSEACENPAC) or the Naval Sea Support Center, Atlantic (NAVSEACENLANT), as appropriate.

231-7.3.4 INFORMATION TO INCLUDE IN REPORT. The report shall include information as appropriate and applicable to the particular turbine. Use a copy of Figure 231-7-3 or similar form. Information required is:

a. Turbine identification, such as number 1, 2, 3, 4 (or port, starboard, inboard, or outboard), high pressure (HP), HP or intermediate pressure (IP), low pressure (LP), main turbine (MT), or SSTG

b. Turbine manufacturer and serial number (of rotor and casing where different)

c. Applicable turbine technical manual

d. Month and year of last turbine casing disassembly and any ship alteration (SHIPALT) accomplished inside the casing.

e. Approximate number of turbine operating hours since the date specified in item d.

f. As applicable to the particular turbine (refer to the turbine technical manual):

1 Rotor position indicator readings taken and date

2 Indicated thrust bearing clearance (inches), design (inches), and date

3 Blade shroud clearance taper gage reading taken (date). Blade shroud clearance (inches), design (inches).

4 Rotor position by reference measurement taken (date) in accordance with figure (number) in turbine technical manual. Measured (inches), constant (inches)

g. Statement concerning radial position of turbine rotor as determined by bridge gage readings, bearing depth micrometer readings, bearing shell (crown) thickness readings, or combinations of these readings and date

h. Statement concerning the condition of bearing journals and magnitude of bearing clearances and date inspected

i. Statement concerning the condition of blading and other parts seen during the last visual inspection through access openings. Include statement about erosion, corrosion, and any damage due to passage of foreign material and date inspected.

j. Statement concerning any casualties affecting the turbine since the last report and date. Include any loss of lube oil. If a boiler carryover is known or suspected to have occurred while the turbine was receiving steam from that boiler, state it.

k. Statement concerning any known or suspected turbine problems, such as high- or low-stage pressures, failure to make required full-power propeller rpm, excess gland steam leakage, vibration, casing cracks or casing joint leak. If casing horizontal joint sealing grooves have been filled with compound, state it. Include date of problem.

l. Recommendation by the Commanding Officer that the turbine casing should or should not be disassembled for inspection and overhaul of the turbine during the next regular overhaul.

TURBINE HISTORY

USS______________ Hull______________ Date______________

a. Identification____________________ b. Turbine Manufacturer______________

Serial Number____________________ c. Technical Manual No.______________

d. Casing Date____________________ Ship Alterations____________________

e. Operating Hours____________________

f.1 Rotor Position Indicator____________ Date______________

f.2 Thrust Bearing Clearance____________Design______________Date__________

f.3 Blade Shroud Taper Gage____________ Design______________Date__________

f.4 Thrust Float______________ Design______________Date__________

g. Radial Position of Rotor______________________________ Date______________

h. Condition of Bearing Journals__________________________Date______________

Journal Bearing Clearance______________________________Date______________

i. Blading____ Date______________

j. Casualties____Date______________

k. Problems____ Date______________

l. Recommendations____________________________________ Date______________

Remarks____

____

____

____

____

Note: Refer to paragraph 231–7.3.4 for specific details.

Figure 231-7-3 Turbine History

231-7.3.5 ACTION TO BE TAKEN ON REPORT. The Type Commander shall decide on casing disassembly (paragraphs 231-6.1.3 through 231-6.13.3).

231-7.4 REPORT CONCERNING OVERHAUL OF TURBINE

231-7.4.1 RESPONSIBILITY FOR SUBMITTING REPORT. When a turbine is disassembled for overhaul, a report shall be prepared on turbine condition revealed when opened, repairs made, and condition when closed. Many turbine technical manuals contain an overhaul report form that should be used. The activity performing the work (such as a U.S. Navy shipyard, repair facility, or ship repair facility under the cognizance of a Supervisor of Shipbuilding Conversion, and Repair (SUPSHIP) USN, Navy Repair Ship, or tender) shall submit the report. If lifted by the ship's force or by any activity other than those listed in this paragraph, the ship Commanding Officer shall ensure that the report is submitted.

231-7.4.2 INFORMATION TO INCLUDE IN REPORT. The report shall describe significant damage noted in the inspection. Parts requiring inspection and conditions to look for are listed in Table 231-7-2. The report shall include a table of all important turbine measurements and clearances before and after repairs. Report accomplishment of any SHIPALT.

231-7.4.2.1 If any repair work is done that affects the suitability of standard repair parts, send a copy of the report to the Naval Inventory Control Point (NAVICP) Fleet Material Supply Office.

231-7.4.2.2 Forward the turbine overhaul (or closing) report to SEA 03Z23, the ship concerned, and the ship's planning yard.

231-7.5 IDENTIFYING CORROSION AND EROSION FOR REPORTING

231-7.5.1 GENERAL. Corrosion and erosion are two types of material attack that can produce damage severe enough to require replacement of expensive turbine parts in a relatively short time. Certain types of erosion and corrosion take place typically in specific areas in a propulsion turbine. The cause and location of corrosion are discussed in paragraphs 231-7.5.2 through 231-7.5.2.3. Erosion is discussed in paragraphs 231-7.5.4 through 231-7.5.4.3.

231-7.5.2 CORROSION. Corrosion is an electrochemical action. To produce corrosion, it is necessary to have:

a. An anode

b. A cathode

c. An electrolyte

d. An external electron path

231-7.5.2.1 Electrochemical Corrosion. The anode and cathode can be two dissimilar metals or the same metal with local areas of differing potentials. The electrolyte may be any form of moisture. The external electron path usually exists as a direct contact between the anode and cathode. This electrochemical action causes destructive removal of metal at the anodic areas where the metallic ion enters the solution.

231-7.5.2.2 Cause of Corrosion in Turbines. The common type of red rust that is characterized by flaking can be a problem with plain carbon steel casing portions if proper drying procedures are not followed during shutdown (Section 9).

231-7.5.2.2.1 During shutdown the combination of boiler carryover deposits and moisture can produce such pitting of blading, wheel disks, and diaphragms that blading edges are completely destroyed and blading vane sections develop holes. This type of corrosion is normally seen in superheated stages of turbines being supplied with steam from an integral superheater-type boiler. This arrangement precludes operation with saturated steam, eliminating the beneficial effect of the dissolving and washing away of these carryover deposits during normal securing and startup routines.

Table 231-7-2. INSPECTION GUIDE

Turbine Part	Look For
Interior surface casing	Corrosion, erosion, condition of casing steam seal surfaces. Take samples of any deposits. Steam cutting or leak paths in horizontal joint and steam chest joint. Cracks in casing, particularly in first-stage shell area, and floor of main steam chest. Photograph any unusual damage or missing parts.
Nozzle diaphragms	Erosion or corrosion of horizontal parting and vertical steam seal surfaces, condition of radial or axial crush pins. Erosion or corrosion, pitting, cracking of vanes.
Diaphragm and gland labyrinth packing rings	Wear, freedom of movement, condition of springs, position relative to rotor.
Valves and seats	Steam cutting of seating surface and integrity of stellite inserts. Cracking of seat seal welds or expansion lips.
Rotor blading and shrouding	Cracks, tears, erosion or corrosion of more than 10% of the cross-section area of the blades, lifting of shroud, integrity of stellite shields, axial and radial rubs, blades, shrouds, or tenons missing.
Rotor	Balance weights intact and firmly anchored (record positions). Erosion or corrosion of packing areas and wheel water cutting. Integrity of end plugs on hollow rotors. Magnetic particle test per MIL-STD-271.
	Journal condition (scoring, wear, pitting), measure journal diameter, thrust collar (scoring, wear, pitting) (see Table 231-6-4).
Journal bearing	Wear, loose babbitt, contact pattern, scoring, tin oxide, RTE intact and operational.
Thrust bearing	Shoe wear, loose babbitt, contact pattern, scoring, wear of leveling plate supports, RTE intact and operational.
Blading radial seals	Wear, pieces broken out, firmly seated.
Valves (nozzles, bypass, transfer, extraction, and drain)	Scoring of lift rods in way of bushings. Conditions of stems that may indicate leakage or hangup. Wear of linkages, bearings, cams.
Rotor-position differential expansion indicators	Wear, adjustment, calibration.
Nozzle blocks	Cracks in ligaments between reamed nozzles. Bolt retainer plates out of position. Loose or broken bolts.
Miscellaneous internal fasteners	Locked properly. Loose or broken bolts.
Steam strainer	Foreign material, damage to basket

231-7.5.2.3 Descriptive Terms Used in Reporting Extent of Corrosion. The degradation shown in Figure 231-7-4 is an example of carryover-induced corrosion attack. It is considered an advanced and severe case. Although most of the surface of the blading and about half of the wheel surface have been affected, the particular row would not be considered in immediate danger of mechanical failure since steam loadings are low and the row is unsusceptible to resonant fatigue failure. Further attrition of blade metal, which would introduce the element of failure from loading, would be termed extreme. Earlier stages of this type of corrosion attack have been produced in laboratory tests. The blading condition shown in Figure 231-7-5 would be termed very light, and that shown in Figure 231-7-6 would be termed moderate to heavy. Other terms such as heavy and light to moderate (as appropriate) will permit establishing various graduations of severity, with the figures in this section as bench marks. Since this type of corrosion attack occurs in an area that is normally seen only with the casing cover lifted, it would be both opportune and prudent to replace parts where corrosion is heavy.

231-7.5.3 WATER SLUGS. A water slug can enter a turbine from an improperly drained main steam system or from boiler carryover. A severe water slug reduces turbine speed, increases thrust-bearing loading, and causes casing and rotor distortions, which usually results in vibration. The seriousness of the vibration, the duration of the slug, and the condition of the thrust bearing from the additional loading all affect the course of action to be taken. As a minimum, slow the turbine until operators are satisfied that the rotor position is proper (thrust bearing intact) and that vibrations are not excessive. Record carryover and water chemistry in the Engineering Log.

231-7.5.4 EROSION. Erosion is similar to corrosion in that it is characterized by attrition of metal surfaces. Here the similarity ends since corrosion is electrochemical in action while erosion is purely mechanical wear. Turbine erosion occurs in areas where water or water particle velocities are high. Turbine blading damage from erosion is usually limited to areas near the blade nose or tips of the last few rotating rows of low-pressure (LP) turbines. Damage is incurred by the moving blade running into the slower moisture particles formed in these wet stages. Such erosion is characterized by an essentially arrested condition after an initially active period. Stellite shielding applied to susceptible areas has largely eliminated blading damage from this source. Blade replacement as a result of erosion is not recommended. Obtain technical assistance as needed.

231-7.5.4.1 Wet-Steam Erosion. Structural elements in LP turbine exhausts can also be eroded by wet steam, but the rate of deterioration is slow and, as with blading, highly susceptible areas are protected by sleeving of erosion-resistant metals.

231-7.5.4.2 Steam Cutting. Steam cutting, also an erosion phenomenon, is identified with turbine parts forming steam seal surfaces. Such metal surfaces, which permit steam to leak from marginal or otherwise inadequate sealing forces, are cut in the direction of the leakage steam flow. Although the examples of this erosion shown in Figure 231-7-7 and Figure 231-7-8 are considered severe, structural integrity is not significantly affected. Leakages at diaphragm parting faces (Figure 231-7-7) and the horizontal joint (Figure 231-7-8), however, are of a magnitude to reduce turbine power output significantly. Typical steam cutting of a poppet valve at stellite inlays is shown in Figure 231-7-9.

231-7.5.4.3 Reporting Erosion. As in corrosion, it is important that sufficient information be given in an erosion report so that recipients can judge severity. In the report, state the part(s) of the turbine affected (part identified preferably by technical manual nomenclature), the appearance of the surface described, and estimates of the depth of attack and percentage of surface affected. Include where possible, photographs showing range of damage, worst to least, to supplement verbal descriptions. Special techniques of erosion repair are discussed in paragraphs 231-8.18 through 231-8.18.2.3.

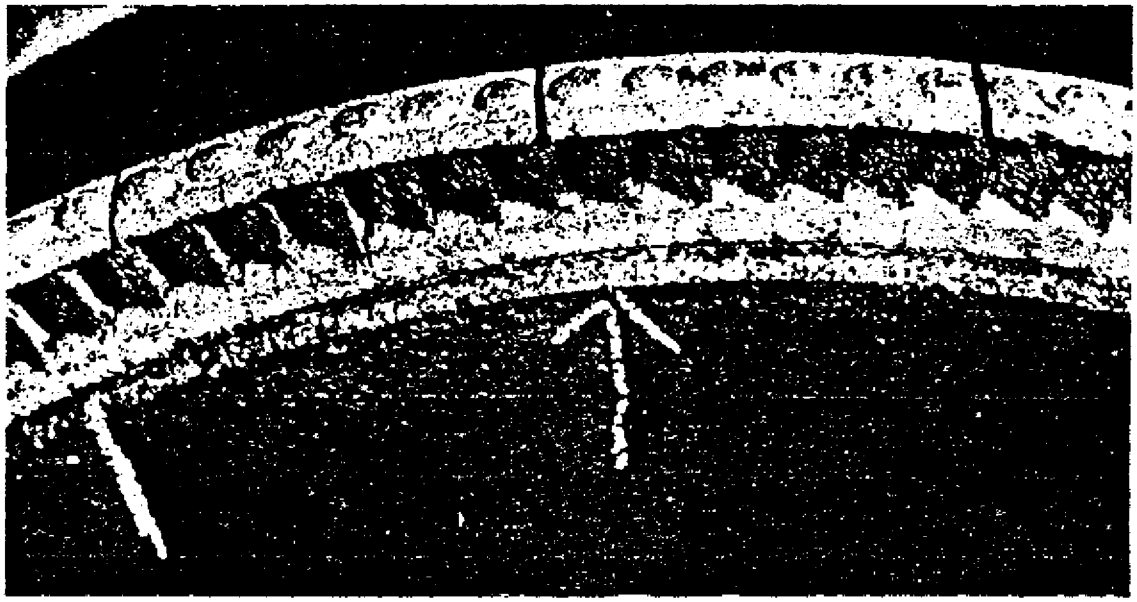


Figure 231-7-4 Extreme Corrosion

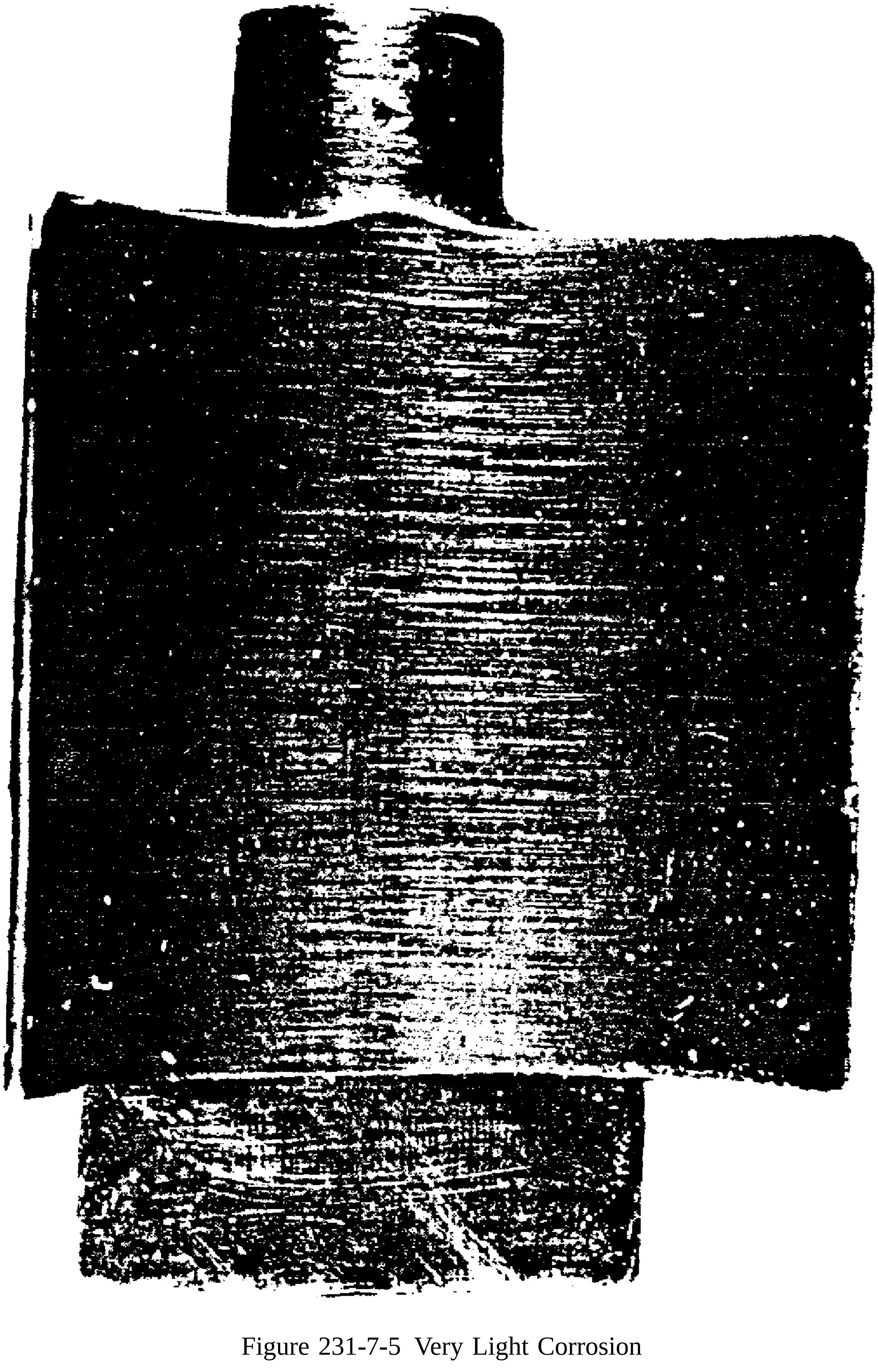


Figure 231-7-5 Very Light Corrosion

Figure 231-7-6 Moderate-to-Heavy Corrosion

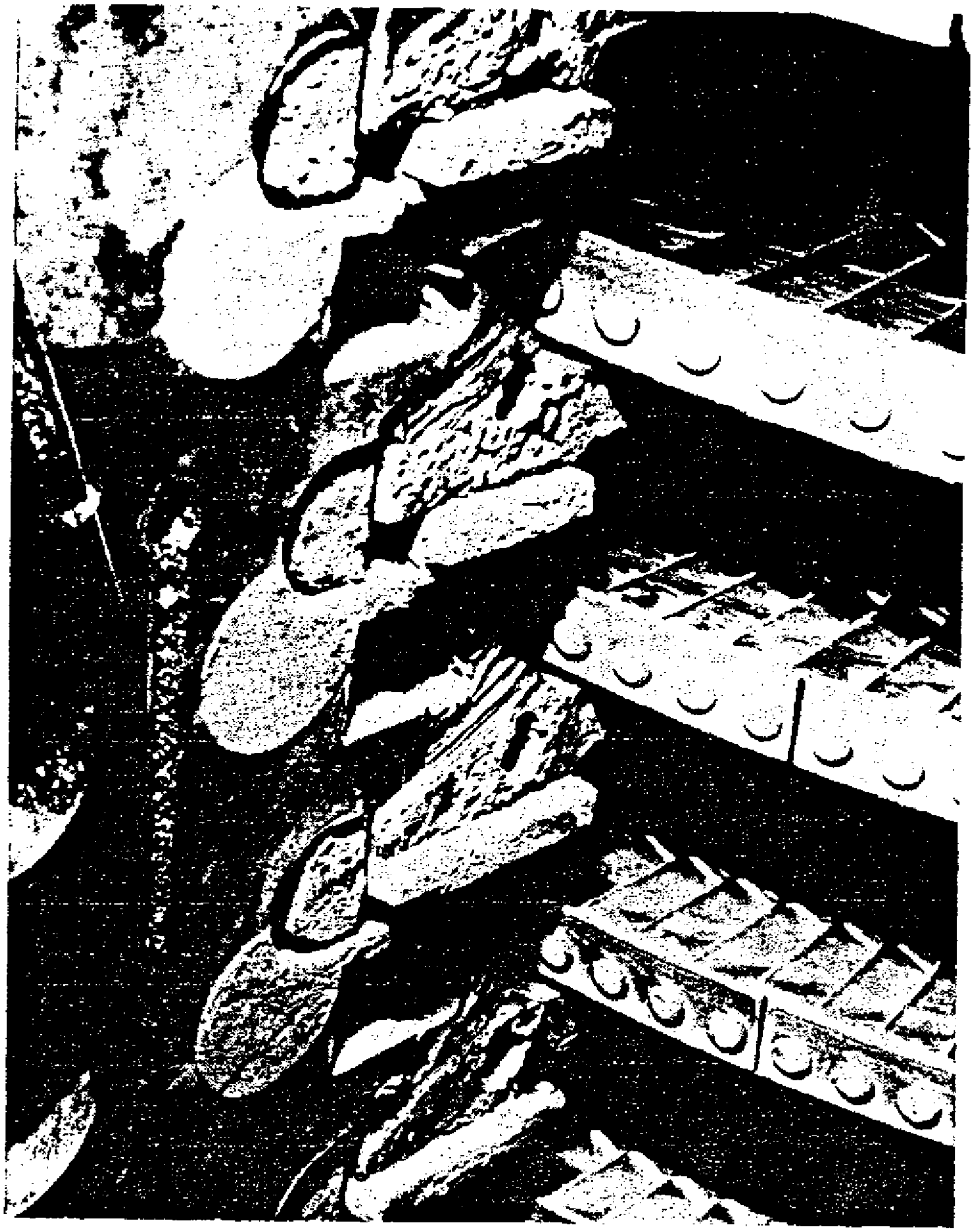


Figure 231-7-7 Diaphragm Parting Face Leakage

Figure 231-7-8 Horizontal Joint Leakage

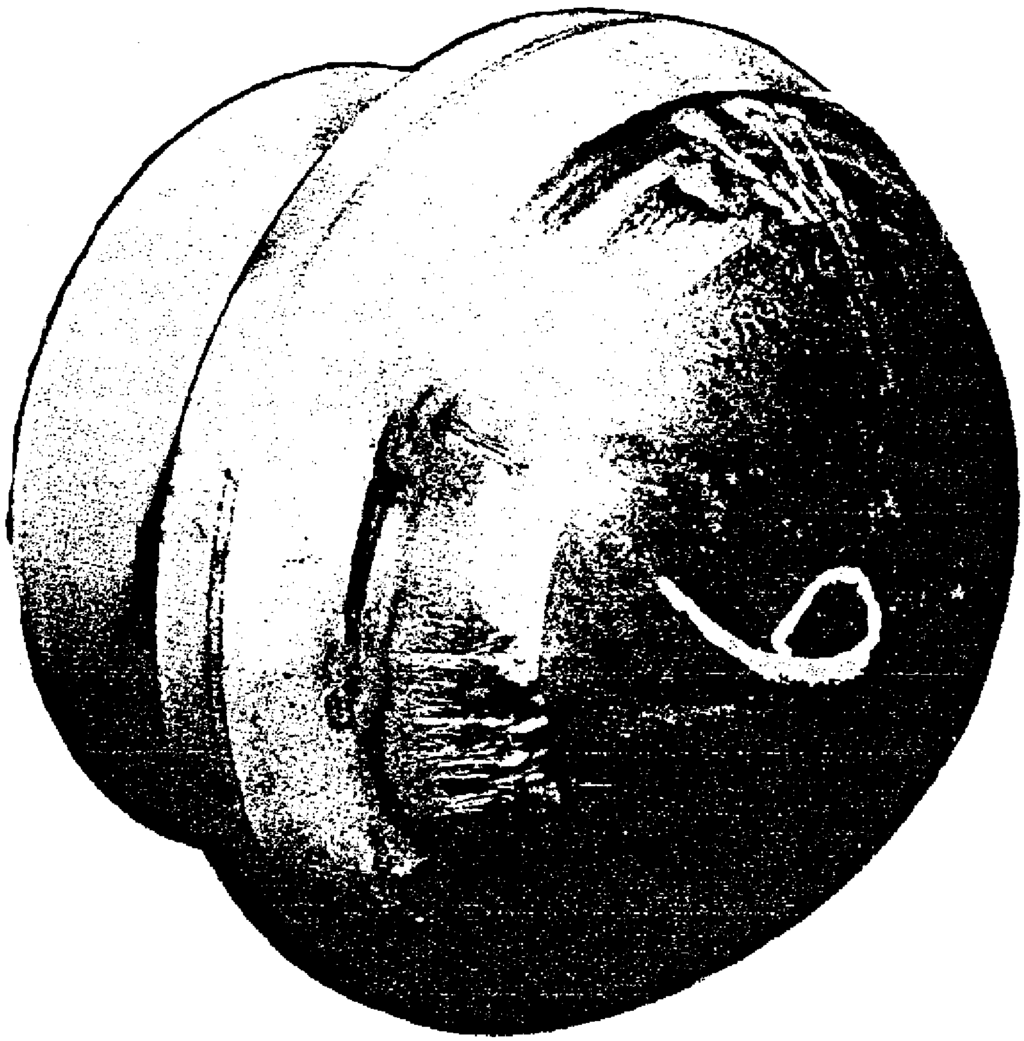


Figure 231-7-9 Steam Cutting of Poppet Valve at Stellite Insert

SECTION 8
REPAIR PROCEDURES

231-8.1 GENERAL

231-8.1.1 APPROVED REPAIR PROCEDURES. Through the years, certain procedures have been developed for repairing propulsion and ship service turbine generator (SSTG) turbines. This section presents those procedures and covers those repairs that are considered to be of major importance. The need for repair may result from a malfunction or may be disclosed during inspections. The procedures cover restoring to working order parts that are considered to be adequately designed. Recommendations on design changes (material or geometry) are welcomed, but to maintain configuration control, such modifications are authorized only through approved ship alteration (SHIPALT). Restrictions and process requirements regarding welding can be found in NSTM Chapter 074, Volume 1, Welding and Allied Processes.

231-8.1.1.1 Technique Improvements. The repair procedures described are considered existing practice. Repair activities are encouraged to modify or substitute alternative techniques that will produce an equivalent repair at reduced cost or a more efficient repair at same cost. Details of improved techniques should be submitted to the Naval Sea Systems Command (NAVSEA) for evaluation and incorporation into this chapter, if appropriate.

231-8.1.1.2 Class B Overhaul. Unless otherwise required and specified, steam turbines and components being repaired or overhauled shall be repaired or overhauled to a Class B overhaul condition. A Class B overhaul is defined by OPNAV Instruction 4700.7F as restoring the operating and performance characteristics of a system, subsystem, or component to its original design and technical specifications. SHIPALT's and modifications, even if applicable, are not to be made unless specified by the customer. It is not required that the appearance of the end product be in ″like new″ condition. The repair activity will demonstrate or guarantee that the end product successfully meets all performance criteria specified by the governing specifications.

231-8.1.2 STEAM PLANT CLEANLINESS. On nuclear powered ships, the use of chemicals, solvents, and compounds in steam turbines, as described in the following paragraphs, shall comply with NAVSEAINST 9210.36, Steam Plant Cleanliness or NAVSEA 0989-064-3000, Cleanliness Requirements for Nuclear Propulsion Plant Maintenance by Forces Afloat, whichever is applicable. The internals of the turbine shall be isolated, the work area vacuumed, and the final wiping done with pure water, denatured alcohol, or other approved solvent.

231-8.1.3 STEAM CHEST CONFIGURATIONS. A typical steam chest valve seat area configuration is shown in Figure 231-8-1. Actual configurations vary slightly among manufacturers and are shown in turbine technical manuals or detail drawings. A steam chest is that part of the turbine upper casing or high-pressure head that encloses the steam control valves and valve seats. Steam chests in propulsion turbines and SSTG turbines have configurations similar to Figure 231-8-1, usually with multiple (typically five or six) valves and valve seats. Some steam chests have casing grooves, and some do not. Propulsion turbine astern valves or SSTG trip throttle valves also have configurations similar to Figure 231-8-1, but with only one valve seat. They have valve bodies instead of steam chests, but the general repair procedures described here also apply to them. Metals used in steam chest (or valve body) castings include carbon steel, 12 percent chrome steel, and chrome moly steel. Some steam chests have Inconel inlays at the valve seat lip, and some do not. Valve seat materials include chrome moly, 12 percent chrome steel, Inconel, or 12 percent chrome steel with Inconel inlay at the valve seat lip. Repair activities are responsible for determining these base materials and their boundaries before machining or welding.

231-8.2 VALVE AND SEAT REPAIR

231-8.2.1 DESCRIPTION OF VALVE AND SEAT Turbine throttle valves operated by individual cam- or bar-lifts have valve disks with hemispherically shaped seating areas. New valve seats are also machined to a spherical radius or conical form at the seating line. Each disk and seat has an overlay of stellite hard facing in seating contact areas. Spherical valve design enables the valve to seat properly even with slight misalignment of disk and seat centerlines caused by uneven expansion of the valve and steam chest assemblies. A valve disk, or poppet valve, is shown in Figure 231-7-9.

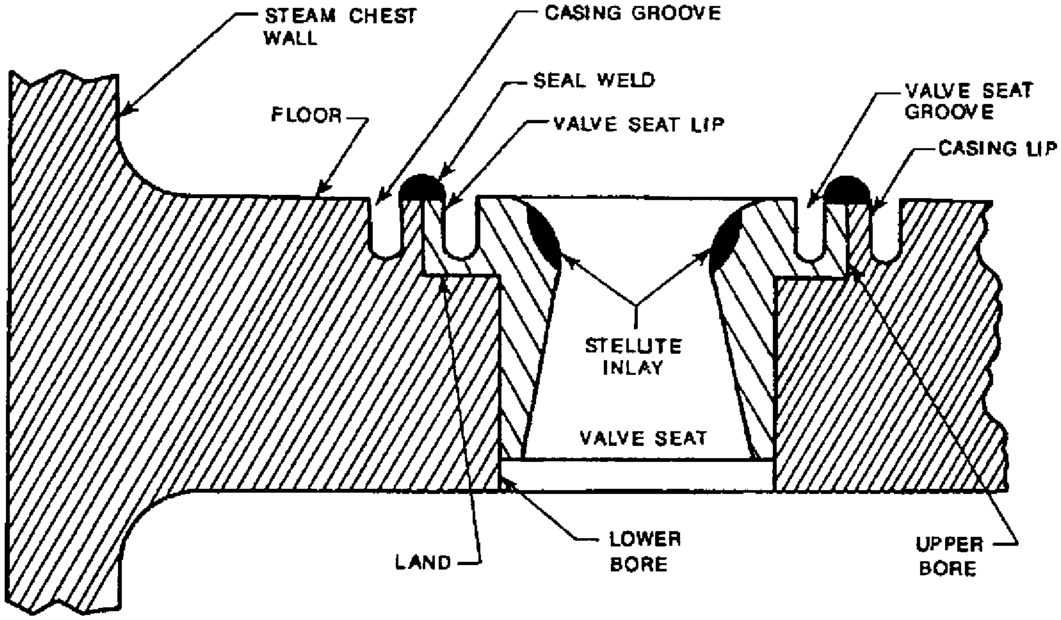


Figure 231-8-1 Steam Chest Terminology

231-8.2.2 VALVE REPAIR FREQUENCY. Different classes of ships have different maintenance philosophies or plans for when to repair valves, and these maintenance plans should be followed. In general, valves should be repaired on the basis of time since last periodic repair or when they fail operational testing, whichever comes first. A periodic repair, without regard to an operational test failure, should be performed when the ship is in a major availability of approximately a year at a depot level shipyard qualified to do the detailed inspections and possible restoration to design conditions that are not normally possible at an Intermediate Maintenance Activity (IMA) such as a tender. The time periods between these periodic valve repairs vary on the basis of the ship class maintenance schedule. A valve repair due to operational test failure can be done at either a depot or an IMA facility, as convenient.

231-8.2.3 TESTING CONTROL VALVES OPERATIONALLY. Test control valves operationally as follows:

a. Control valves are subjected to erosion, particularly under conditions of severe wet-steam throttling. Steam cutting makes repair necessary at more frequent intervals than the overhaul cycle.

b. Because turbine sections are subjected to great cyclical temperature changes over a period of many years, they can develop cracks. The steam chest is a susceptible area. Cracks most often appear at the valve-seat expansion ring seal weld and less frequently at the juncture of the chest floor and vertical walls. Either may leak steam in a manner that bypasses the closed valves and, under extreme conditions, may leak enough steam to keep the shaft turning after the throttle valve has been closed.

c. In the course of operating, there are three ways to judge the tightness of these valves at shutoff: listening for leakage at the steam chest or at the astern throttle valve or SSTG trip throttle valve, observing an increase in rotor coastdown time, and observing a pressure drop in an isolated steam header. Using a listening rod or stethoscope at the chest area can improve test sensitivity when background and normal engine room noises are high. Sonic leak detectors have been used with some success.

d. For propulsion plants with clutches, leakage that sustains rotation of turbines and of unclutched gears with the throttle valve closed is evidence of valve seat or disk erosion. Note the condition in engineering logs and records, but corrective action is not mandatory at this stage. Main turbine leakage that overcomes breakaway torque and rolls turbines and gears from a standstill is dangerous enough to warrant immediate correction.

e. For ship propulsion units without clutches, leakage that can sustain rotation of turbines, gears, shafting, and propeller with throttle closed is cause for concern. Schedule corrective action. When steam leakage with throttle closed overcomes breakaway torque and rolls turbines from a standstill, corrective action is mandatory.

f. Leakage testing of SSTG trip throttle valves can be conducted at normal operational steam pressures and operational vacuum. With governor valves open and an operator ready to shut them, trip the trip throttle and verify that the SSTG decelerates and coasts to a complete stop. Do not allow acceleration or overspeed conditions. Specific testing, if available in the technical manual or Planned Maintenance System (PMS) for a specific trip throttle valve, takes precedence.

g. Leakage testing of SSTG governor valves can be conducted at normal operational steam pressures and operational vacuum with SSTG unloaded and without field flashed. With an operator ready to trip the trip throttle valve, verify that the governor valves can maintain the SSTG speed at the low-speed stop or minimum speed settings. Do not allow acceleration or overspeed conditions. Specific testing, if available in the technical manual or PMS for a specific governor valve, takes precedence.

h. When a turbine continues rolling with the valves shut, verify proper valve adjustment and proper gland seal operation before repairing control valves.

231-8.2.4 VALVE INSPECTION AND REPAIR PROCEDURE. If valve seat or steam chest inspection or repair is required as a result of failing operational testing or required periodic inspection and repair, proceed according to the following paragraphs.

231-8.2.4.1 Preparation. Obtain the proper ship’s authorization and tag out. Assign qualified mechanics who have a history of successful valve work for the full duration of the repair.

231-8.2.4.2 Disassembly. Remove valve chest cover, or valve bonnets, and nozzle-control valves for access to valve seats and chest internals. See turbine technical manual for specific disassembly procedures. Document the as-arrived valve settings.

231-8.2.4.3 Caps and Plugs. Work can be done in the chest without lifting the main casing upper cover, but take all reasonable precautions to avoid entry of foreign material through the valve seats. A cap or plug should be constructed to cover the valve seat opening. A cap that can be locally manufactured out of sheet metal and installed is shown in Figure 231-8-2a. A plug that can also be manufactured locally and installed is shown in Figure 231-8-2b. Pearl Harbor Naval Shipyard drawings such as 938-185-83NT apply. When necessary, flexible foam cleanliness plugs may be used. Whenever foam is used, check the foam before each use to ensure it is a solid piece and not shedding or sloughing off particles. Take action to ensure that the foam plug will not fall through the valve seat. Tape may be used to secure an acceptable type of plug or cover in place or to seal around the edges of an acceptable type of plug or cover. For nonnuclear ships, a clean, hemmed cloth can be inserted into the steam passages beneath the valve seats, but the cloth should be fitted to completely seal off the opening.

231-8.2.4.4 Steam Chest Inspection. Visually inspect the steam chest walls and floor, not in valve seat area, to verify no cracks or pits. Nondestructive Testing (NDT), such as liquid penetrant testing, is not recommended because of the irregular surface of a cast steam chest. NDT may be used to investigate visual indications, if necessary. Visual cracks and pits in a cast, not fabricated steam chest, may be ground out and blended with surface contour without weld repair if they are:

a. 1/4 inch deep or less or

b. 20 percent or less of design casting thickness, whichever is less.

231-8.2.4.4.1 Stop grinding out or chasing any crack or pit that grows in length or width as it is excavated. Report to NAVSEA and the manufacturer these casting faults and those that remain after allowable grinding. Do not weld on any steam chest, except for valve seat, bushing seal welds, or carbon steel welding, without NAVSEA approval.

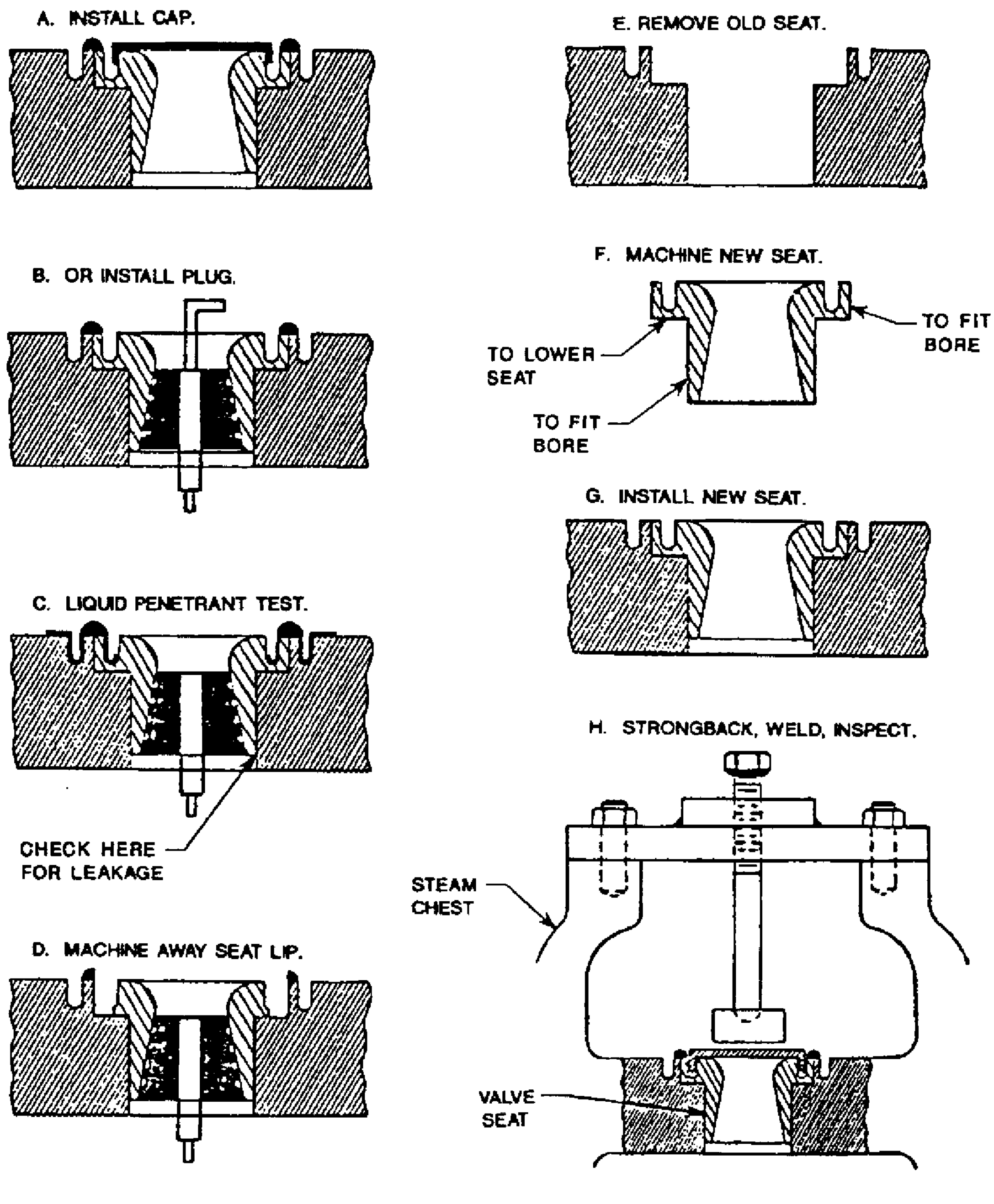


Figure 231-8-2 Valve Seat Replacement

231-8.2.4.4.2 Inspect the steam chest cover visually as described in paragraph 231-8.2.4.4. Inspect lift rod and bushing for proper clearances and condition. Lift rod bushings that are removed will often gall the lift rod bushing bore, making cleanup machining necessary and enlarging the bore. Use of replacement bushings with oversized outside diameters is allowed with local engineering direction and without a NAVSEA waiver. Before establishing lift rod bushing parallelism, verify that the steam chest cover flange is not bowed. Do not install a nonhardened bushing where a hardened bushing is required or where galling with the lift rod may seize the controls. For some carbon steel covers, SHIPALT's exist for cladding the lift rod bores with corrosion resistant alloy and machining to size. Inspect turbine control linkages, moving joints, pins, and pin bushings for proper condition and clearances. Existing pins and pin bushings may be replaced with hardened bushings and normal pins with local activity engineering approval. Thus, future maintenance will involve pin replacement only. Turbine controls should be reassembled per detail drawings and set to the original equipment manufacturer's (OEM's) control drawing instructions.

231-8.2.4.4.3 Some steam chest covers have trepan grooves machined around the bushing bores. The resulting lip is rolled over the boundary of the bore to retain the bushing. This trepan material is removed when the bushing is being removed and usually cannot be used again to trepan the new replacement bushing. A replacement bushing may be retained by weld prepping and seal welding (using a qualified procedure) to the steam chest cover bushing bore. The worker shall qualify on a mock-up. See paragraph 231-8.3.4 for guidance.

231-8.2.4.5 Expansion Groove Inspection. Inspect casing and valve seat grooves and lips and seal welds for drawing dimensions, and verify lack of visible cracks (Figure 231-8-1). If seal welds bridge or flow into a casing or valve seat groove, the expansion feature will be lost. Excess weld can be carefully machined away, or the valve seat replaced. Valve seat lips or casing lips that have been machined away must be reestablished to drawing dimensions. Inconel lips can be built up with approved welding procedures, or the valve seat replaced. Other casing materials require special welding procedures with postweld heat treatment and NAVSEA approval.

231-8.2.4.6 Valve Disk Inspection. Inspect the valve disk, or poppet valve, seating area for cracks, pits, steam cuts, and surface erosion using paragraph 231-8.2.4.7 on valve seat inspection for guidance. Replace severely steam cut or cracked disks. If no serious defects are present, place the disk in a lathe, and polish the seating area with emery cloth. Check the spherical surface for runout using a dial indicator. The runout in any plane through the seating area and perpendicular to the valve stem should be less than 0.0005 inch. Runouts greater than this but less than 0.005 inch can be corrected by further polishing with emery cloth. If the runout exceeds 0.005 inch, remachine the spherical surface or replace the valve disk. Astern valves and some other turbine valves have double-lift valves where the inner valves cannot meet the above criteria. Do not disassemble these double-lift valves to perform inner seat inspections or blue contact checks. Check that the inner valve lift is correct and that the inner valve does not leak during standing water testing. If double-lift valves must be disassembled, often a trepan groove (see paragraph 231-8.2.4.4.3) is destroyed. If this valve must be reused, it may be reassembled by threading together the outer valve parts and staking them in four places with the proper staking tool. Indents can be machined out to receive the stakes. Inspect the stakes to make sure they have no cracks. Tack welds, made using a qualified procedure, may also be used to secure the outer valve parts.

231-8.2.4.7 Valve Seat Visual Inspection. Inspect the valve seat to determine if the seat is acceptable, if the seat can be lapped, or if replacement is required. With the valve seat throat properly plugged, clean the valve seat to remove discoloration by polishing the metal with fine emery cloth while vacuuming and wiping. At this point the interface between the base metal and stellite overlay and the existing valve to seat line of contact should be visible. A judgment can be made as to whether or not there is sufficient overlay metal remaining to support removal of surface defects. The stellite seating area, especially on the line of contact, should be checked visually for cracks. If open imperfections, cracks, steam cuts, wiredrawing, pits, or erosion that can be felt with a sharp machinist's scribe are found, lap the valve seat or replace. Experience has shown that open imperfections of 1/64 (0.016) inch or deeper cannot be readily lapped away. Activities can make this decision on the basis of their own experience and their own success history at lapping versus replacement. Tight hairline cracks in the stellite that are visible but not detectable by feel are acceptable.

231-8.2.4.8 Valve Seat Liquid Penetrant Testing. Steam chest casting near the valve seats and valve-seat welds shall be tested for cracks by liquid penetrant testing (Figure 231-8-2c). The liquid penetrant material shall be in accordance with MIL-I-25135, group 1, Inspection Materials, Penetrants. The test procedures, which include surface preparation and pretest cleaning and inspection techniques, shall be in accordance with MIL-STD-271, Requirements for Nondestructive Testing Methods. All tests shall be performed by qualified personnel in accordance with NSTM Chapter 074, Volume 2, Nondestructive Testing of Metals, Qualification and Certification Requirements for Naval Personnel (Non-Nuclear).

231-8.2.4.8.1 The following test procedure should be performed to detect cracking and excessive porosity of valve seat and casting material that could result in steam leakage:

a. Liquid penetrant test the machined chest casting near the valve seats, the seal weld, the casing groove, and the valve seat groove shown in Figure 231-8-2c.
b. Grind out any linear or nonlinear indications found during the liquid penetrant test. (See paragraph 231-8.2.4.4 for guidance.) If indications are in a valve seat, the seat can be replaced.
c. Perform the following dye puddle test to determine if there are cracks or porosities that form a leak path past the steam chest floor. Such defects can cause turbine roll-over problems.
   1 Using an eyedropper, carefully fill the casing expansion groove with penetrant material to approximately 1/8 inch below the seal weld.
   2 If no immediate leakage is noted below the valve seat (Figure 231-8-2c), allow the penetrant to remain in the groove for 4 hours.
   3 If the casing expansion groove shows no leakage after 4 hours, fill the valve seat expansion groove with penetrant and test in the same manner. This test, however, may be discontinued after 2 hours because of the shorter leakage path involved.
d. Thoroughly clean the area after the leak test with solvent recommended by the penetrant manufacturer. First, remove the penetrant material with absorbent paper, wipe dry, and then remove with solvent until no evidence of liquid (dye) remains in the crevices.

231-8.3 VALVE SEAT REPLACEMENT

231-8.3.1 GENERAL. If a valve seat inspection (paragraph 231-8.2.4.7) indicates that valves cannot be lapped satisfactorily, replace the seats. With caps or plugs installed as described in paragraph 231-8.2.4.3, machine away the seal weld (Figure 231-8-2d). Take extreme care not to machine the steam chest casing. In this preferred way the inner edge of the seal weld is removed along with much of the lip of the seat, but the casing lip remains intact. If this cannot be done, it is also possible to machine away the seal weld carefully until the crack between the seat lip and casing lip appears full circle. The problem with this second technique is that some of the casing lip is usually removed. If this casing lip were part of an Inconel inlay, it could be buttered or weld built up with Inconel and remachined to drawing dimensions before installing the valve seat. If it were made of 12 percent chromium, however, it would need to be built up with 12 percent chromium (or with Inconel with a special wavier), requiring a special weld procedure with postweld heat treatment stress relief, which can cause casing distortion. Use of hand tools, hand-held grinders, cutting torches, or similar equipment is prohibited because damage to the steam chest could prevent a successful repair. An exception is allowed if the machinist is qualified (paragraph 231-6.1.3.3) to perform a local procedure for hand-grinding in this application. Machine tools are valuable for removing valve seats. They mount on the steam chest flange and are precisely controllable. One is Grimsley Machine No. VBB-850 by Grimsley Tools, 524 Middle St., Portsmouth, VA, 23704, (804) 309-4438. Another is 206B Bevelmaster (NSN 4319-01-014-9094) by Tri Tool Inc., 3806 Security Park Drive, Rancho Cordova, CA 95742, 800-345-5015. Repair activities are responsible for procuring proper tooling for removing valve seats satisfactorily and efficiently.

231-8.3.2 VALVE SEAT REMOVAL AND BORE INSPECTION. Vacuum metal chips from the steam chest before removing old valve seats. If necessary, seats may be cooled and steam chest warmed to facilitate removal. Pulling tools may be used as necessary if the steam chest will not be damaged. Inspect steam chest bores, shown in Figure 231-8-2e. Verify drawing dimensions of steam chest lip, if any, and upper and lower bore dimensions and land dimensions. Dress surfaces by machining to drawing dimensions to remove burrs and other high metal. Keep metal chips out of turbine and vacuum, as necessary. Some steam cutting and bore irregularities below the seal weld area or below the shoulder in the bore are acceptable, as long as the seat will have adequate support and proper bore clearance. If the bore ends up being too large, investigate oversize seats (paragraph 231-6.17.9) to be machined to size.

231-8.3.3 SEAT FITTING. Fit seats as follows:

a. Machine surfaces of valve seats to fit bore surfaces per manufacturer's detailed drawings. Typically, lower bores should have a slight clearance fit, and upper bores should have a zero to slight clearance fit so that seats may be taken in and out for measurements. Review manufacturer's detail drawings because some valve seats have shrink fits.

b. Machine to adjust the relative heights of the valve seats (Figure 231-8-2f) with respect to each other to minimize the amount of machining adjustments required to obtain the plan valve lift dimensions. Leave enough thickness of metal under the valve seat groove to maintain valve seat integrity.

c. Using a parallel bar on top of the steam chest, take a set of depth micrometer readings 90 degrees apart to check the parallelism of each valve seat to the steam chest flange. Valve seats should be square as specified in the manufacturer's detail drawings, typically within about 1 mil (0.001 inch). This ensures that a successful 360-degree blue contact check is obtainable between the valve and the valve seat. If seat does not sit square, blue check the seat to bore land and machine high spots as necessary, while keeping metal chips out of turbine and vacuuming.

d. Valve seat parallelism is a technique to ensure a successful blue check of valves to valve seats, which is the essential end product. When the specified parallelism cannot be attained either before or after final welding, the repair facility may evaluate the condition to show that it will not prevent a successful blue check and then process a local waiver as appropriate. Note that valve seat parallelism is a tool to ensure a successful blue check between valves and valve seats. When specified parallelism cannot be attained either before or after welding, the repair facility may evaluate the situation, show that it will not prevent a successful blue check, and process a local waiver as appropriate.

231-8.3.4 VALVE SEAT WELDING. The weld from the valve seat lip to the steam chest is a seal weld, sealing the joint from steam leakage and helping to hold the valve seat stationary. Factory seal welding procedures may be adapted by a repair facility and used by qualified welders without further NAVSEA approval. For example, for GE SSN 688CL main and SSTG turbines, the base material is Inconel on both the seat OD lip and the bore ID lip. A Gas Tungsten Arc Welded (GTAW) Tungsten Inert Gas (TIG) seal weld is used with two passes of Inconel filler material (MIL-E-22200/3, MIL-8N12). Preheat is 70°F minimum, interpass is 350°F maximum, and there is no postweld heat treatment. For Westinghouse steam turbine valves, the base materials may be type 309 corrosion-resistant steel (CRES) or Inconel. Use an acid test to determine the base material. Other alloy-identifying instruments may be used, like thermoelectric alloy sorters made by such companies as Koslow Scientific Company, 75 Gorge Road, Edgewater, NJ 07020, 800-556-7569 or Technicorp (NSN 1H-4470-01-282-5133 X5), 646 Eyster Boulevard, Rockledge, FL 32955, 800-541-1971. Use Inconel filler if either base material is Inconel and use type 308 CRES filler if both base materials are stainless steel. Cap the valve seats as shown in Figure 231-8-2h, and use a strongback to securely hold the valve seat bottomed in the steam chest land. Using an approved welding procedure and a qualified welder, seal-weld the valve seat to the steam chest. First, tack-weld the seats in place, and then verify parallelism, as specified in the detail drawings (typically within 3 mils), before actual seal welding. If a valve seat is found lifted or cocked after tack welding or seal welding, then the weld is acting like a structural weld and is subject to cracking. The valve seat should be removed and correctly seal welded, bottomed flush to the steam chest land.

231-8.3.5 RETESTING VALVE SEATS. Perform the inspection and NDT specified in the welding procedure, but as a minimum there should be liquid penetrant testing with no cracks or porosity. If found, they may be weld-repaired with the same procedure. The line of junction, undercutting, between the seal weld and steam chest or valve seat will sometimes hold liquid and should not be mistaken for a linear indication. Verify that seats have not lifted during welding and have kept their parallelism, as specified in the detail drawings (typically no greater than 3 mils), to the steam chest cover to facilitate a good contact check. Hydrotesting of steam chests is not required, and liquid penetrant testing has been shown to be sufficient.

231-8.3.6 VALVE-TO-VALVE-SEAT CONTACT. The procedure to verify valve-to-valve-seat contact is as follows:

1. Verify that valve seat is plugged.
2. Place a small amount of Hi-Spot Prussian Blue on the valve disk and smooth out. This coating should be light and can be achieved by wiping the disk at this point with a cloth.
3. Place the lift beam assembly in the steam chest and rock the valves slightly (a couple of degrees off the vertical centerline) while pressing down on the valves. Do not rotate valves.
4. Remove the lift beam assembly and check the valve seat for contact. Contact should be a continuous line and shall be 100 percent. If 100 percent contact is not achieved, then lap and recheck the seat until 100 percent contact is obtained.
5. Clean with a vacuum cleaner and a rag with approved solvent.

231-8.3.7 RESURFACING VALVE SEATS. Valve seats need resurfacing if they do not have a 360-degree line of contact blue check or if they have minor imperfections in the seating area. Valve seat visual inspection is discussed in paragraph 231-8.2.4.7. Do not attempt to lap a seat with a valve. Most General Electric turbine valve seats are resurfaced by lapping a flat angular surface at the line of contact of the valve disk with the seat. The GE SSN688CL main turbine valve seat angle is shown in Figure 231-8-3; for GE SSN688CL SSTG's this angle is 60 degrees. Seats from Delaval and Westinghouse are resurfaced by lapping with a tool that has the same contour as the seat. The stellite overlay is sufficiently thick in new valve seats to permit resurfacing several times before replacement is required. The number of resurfacings depends on the amount of metal removed during each resurfacing, which is a function of valve condition.

1. Resurface valve seats using a cast iron lapping plug and grinding compound. Turn the conical-shaped lapping plug to the same angle as the valve seat angle. The small end of the cone should have a cylindrical turn sized

to fit into the throat of the valve seat and act as a centering guide. Contoured laps should be made to appropriate valve seat drawings. Attach a suitable spindle to the lapping plug to facilitate turning from outside the steam chest.

2. The lapping plug and spindle should be concentric and, if turned between centers, can be easily resurfaced while maintaining consistent geometry. To avoid wobble during lapping, slip a close-fitting guide plate over the spindle and secure at the steam chest flange level. The lap can be driven by any convenient means, the simplest being manually using a sliding T-handle made by putting a small bar through a hole drilled in the spindle.

3. The surface of the lapping plug should be resurfaced often during the grinding operation to ensure correct seating angle and flatness of the valve seat angular surface. Do not grind off any more valve seat material than necessary. After removing all surface defects from the area, check the valve seat for contact with a valve disk of correct geometry using Prussian Blue. The valve seat should have a 360-degree contact with the disk.

4. Other techniques can be developed and used by repair facilities without NAVSEA approval provided they produce a 360-degree line of contact blue check and do not damage the turbine in any other way. Proper tooling is highly recommended as cost effective, and two possible suppliers of valve seat resurfacing tools are:
   a. Leavitt Machine Company
      P.O. Box 270
      Orange, MA 01364-0270
      (617-544-2751)
   b. Unislip Inc.
      424 Wessex Rd.
      Valparaiso, IN 46383
      (801-348-2606)

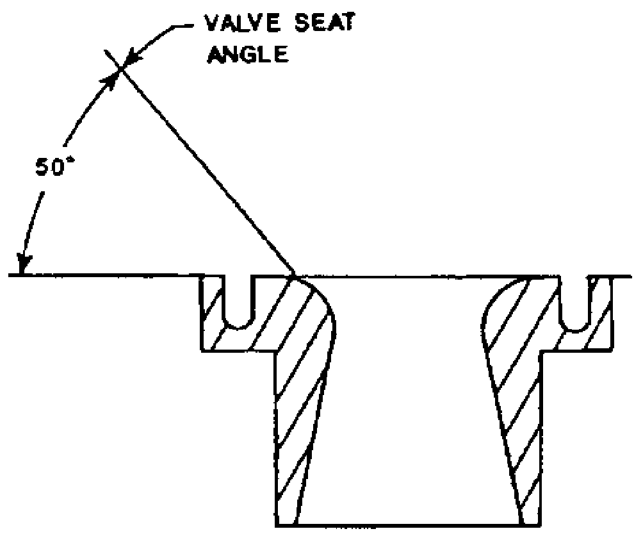


Figure 231-8-3 Valve Seat Angle

231-8.3.8 VALVE SETTINGS. Establish valve beam parallelism within about 3 mils to the steam chest cover. Establish valve hang heights and other valve settings as described in the turbine technical manual or OEM detail drawings.

231-8.3.9 REASSEMBLY. Verify that all tools, plugs, caps, foreign material, and residues are removed from the steam chest or valve body. Reassemble the steam chest joint as described in the turbine technical manual or paragraph 231-8.5.

231-8.4 REPAIR OF LABYRINTH PACKING RINGS

231-8.4.1 SALVAGING WORN PACKING RINGS. Labyrinth packing rings, both gland and interstage, may experience rubs during turbine operation. These rubs, caused by misalinement of the rotor and packing centerlines, either transient or fixed, result in increased packing clearance from reduction in seal tooth height and blunting of seal edges. Such increases in packing clearance, even by amounts greater than specified limits, are not harmful to the mechanical operation of the turbine, except for cases where uncontrollable gland leakage becomes evident. Excessive packing leakage, however, will result in some loss of turbine operating efficiency and economy. Packing clearances should therefore be maintained within the minimum and maximum specified in the manufacturer's instruction book to retain best turbine performance. In cases where no limits are given, the clearances can be allowed to increase to twice the design value without an inordinate effect on performance. Replace packing rings showing excessive clearance. If new packing cannot be procured or manufactured in the time frame available for turbine repairs, the existing rings can usually be remachined to restore design clearances. Check packing springs to ensure that they still provide proper loading.

231-8.4.1.1 Readjusting Fit by Remachining. When necessary, worn labyrinth packing rings can be salvaged by machining a small amount of material from the butt joints and remachining the radial locating surfaces and seal teeth. The surface, dimension C, in Figure 231-8-4 can be reduced to 75 percent of design size. In some cases it may be necessary to replace the amount of metal removed by building up the outside diameter of the ring. This is done by brazing the areas contacting the packing spring to ensure sufficient locating tension.

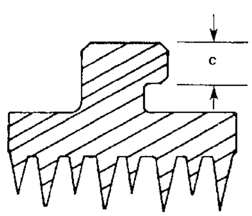


Figure 231-8-4 Packing Ring

231-8.4.1.2 Repairing Damaged Teeth (Fins). To obtain maximum effective sealing, the tip width of packing teeth (fins) should be maintained at 0.010 + 0.005 inch. Resharpen blunted teeth by scraping the sides of the teeth. Either a bearing scraper or cutting tool slotted to the tooth shape can be used. Take care to avoid reduction of tooth height.

231-8.4.2 PROTECTING TEETH. When disassembled, protect labyrinth packing ring teeth (fins) with soft plastic, fabric, cardboard, or other material to prevent damage due to impact or contact.

231-8.5 REPAIR AND MAKING UP OF TURBINE CASING JOINTS

231-8.5.1 IMPORTANCE OF GOOD CONTACT. When making up any steam joint, regardless of the compound being used, the importance of ensuring burr-free, flat surfaces cannot be overemphasized. This is done by first stoning high spots and burrs with a fine bench stone. Pay particular attention to areas around drilled or reamed holes where material is upset. High spots will be indicated by shiny areas, which should be stoned until they begin to blend in. Check both surfaces for flatness with a large (minimum 12 by l8 inches) surface plate.

231-8.5.1.1 Whenever a casing joint is opened for any reason, a final contact check is required and is accomplished by taking a red and blue check of the made-up joint. Contact required is:

a. Seventy-five percent over entire joint plus 1/4-inch minimum continuous contact band inside bolts and across each pressure section.

b. For joints with pumping grooves, same contact required as in item a plus a 1/8-inch minimum continuous contact band inside groove.

c. Certain turbines have different contact areas by design, and the design will take precedence.

231-8.5.2 PROCEDURE FOR CHECKING CONTACT. To check contact:

1. Clean the flange surfaces and pumping grooves (if provided) to remove any previously used sealing compound.

2. Clean and inspect casing bolting in accordance with paragraphs 231-8.7.6.1 and 231-8.7.6.1.1.

3. Apply a thin film of Prussian Blue to the upper casing flange surface. For contrast and easier impression reading, the lower flange may be coated with a thin film of red, chrome yellow, or orange oil-based pigment, which can be purchased in collapsible tubes at local paint stores. Colors too heavily or unevenly applied will appear oily in these locations. Remove excess color by wiping or blotting with a lint-free rag. Spread remaining coloring to a uniform low-luster film. Take care to prevent coatings from entering turbine internals.

4. Reassemble the mating halves of the casing. Lubricate the threads and faces of the nuts and washers, and install all joint bolting. Tighten all studs and bolts in proper sequence to the required blue check bolt torque. This may be done at a low torque (25 percent of final torque) where full contact is a good indication of a proper joint. If a low torque does not give full contact, however, a decision must be made to work the joint surface more or to try up to 100-percent final torque at that time. Check for gaps between joint faces with a feeler gage. Mark any gaps with chalk to establish, for future reference, the length of opening and size of feeler inserted (paragraph 231-8.7.6.3).

5. Remove bolts, and raise upper casing. Observe joint contact shown by color transfer. Large areas showing no contact toward the inner edge of the flange may indicate interference such as incorrect radial clearances at diaphragms or packing housings. If interference is suspected, measure to verify clearances, and repeat the blue check if corrections are made.

6. Poor joint contact resulting from local high spots can be corrected by scraping or stoning the surfaces to remove the high spots. Repair contact patterns showing leakage paths caused by surface erosion or steam cutting by resurfacing the affected areas.

231-8.5.3 REPAIR OF JOINTS. Steam leakage across a turbine casing joint, if allowed to continue for any length of time, will erode the joint surfaces. Erosion damage may range from visible trough-like cuts across the sealing area to slightly rough or pitted areas shown to be low spots by a blue contact check. Low spots up to 0.010 inch in depth can be repaired by scraping and tapering the joint surface. Deep cuts and severely eroded areas, however, require repair welding and resurfacing. Areas less than 0.030 inch in depth may be repaired by electrobrush plating in accordance with MIL-STD-2197, Brush Electroplating on Marine Machinery - Class 2 - Steam Static Sealing Surfaces. Copper and silver plating have been used successfully in the past. Prepare local procedures to comply with MIL-STD-2197 and to exclude chemicals from the turbine interior. Use dams around the area to be plated. Cover and tape the turbine interior as necessary to keep out foreign material.

231-8.5.3.1 Scraping and Tapering. Scraping of steam joints should be done only by personnel qualified by their previous successful scraping experience (see paragraph 231-6.1.3.3). Activities without qualified scrapers should consider contracting out this work. Scrape the low spot or eroded area to restore a suitable sealing surface. The adjacent surfaces should be scraped on a taper to blend the low spot to the surrounding original surface. Both upper- and lower-casing flanges should be scraped if eroded, and the combined taper should not exceed 0.001 inch per inch of flange length. Continue scraping until a blue check shows good contact in the damaged area. Much time and effort can be saved by using a small surface plate, less than 6 inches in diameter, to test and scrape small areas, instead of bolting the casing halves together to determine the progress after each scraping. The use of a surface plate is not intended to make large areas of the flange surface flat and therefore does not eliminate the need for bolted-up flange contact checks during the scraping process for final determination of good joint contact.

231-8.5.3.2 Procedure for Repair Welding and Resurfacing. Prepare the flange surface for welding by chipping a groove approximately 1/8 inch deep to remove the damaged area. The groove should be no less than 1/8 inch wide by 1/8 inch deep, as shown in step 1 of Figure 231-8-5. The sides of the groove should be upset at the joint surface, as shown in step 2. This can be done by hammering with a blunt tool and is necessary since arc welding will undercut the base metal. Weld up the groove, step 3, under the direction of a welding engineer and in accordance with MIL-STD-278, Welding and Casting Standard. To ensure crater-free inlay, tack-weld blocks to the casing so the arc may be struck and broken beyond the joint. Dress off the raised area by filing or grinding, but leave the repaired area a few mils higher than the joint surface. Using a surface plate large enough to span the repaired area and approximately 3 inches of original surface on all sides, scrape the repaired area to blend with the original joint surface.

231-8.5.3.3 Checking Joint for Steam Leaks. Thermal insulation, or lagging, is applied to the casing joints on steam turbines to prevent the joints from becoming much cooler than other parts of the turbine casing, which are also lagged, and thus inducing thermal stresses. The thermal stresses, if large enough, can distort turbine casing joints, which may result in steam leaks to the engine room and possible damage to the turbine. Precautions recommended during inspection of casing joints for steam leaks to minimize thermal stresses in the joint are:

a. All lagging should be in place on the turbine casings and joints before operation on steam.

b. Remove only short sections of lagging (18 inches maximum) for short periods of time (5 minutes maximum). Reinstall each section removed before removing another section.

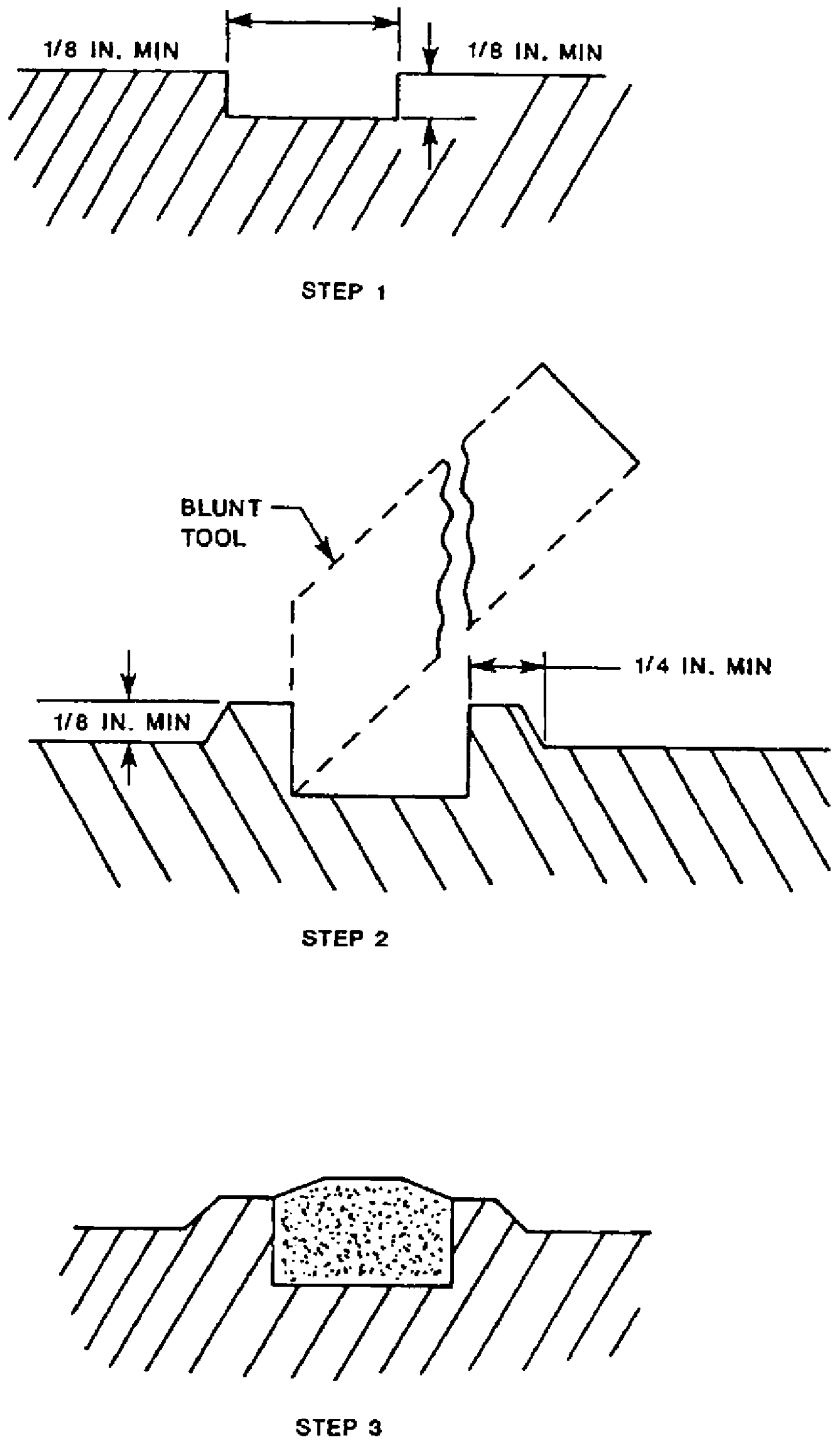


Figure 231-8-5 Flange Surface Repair

231-8.5.3.4 Steam Leak Repairs. Leaking steam joints, whether the leak is a shot of steam or a few bubbles of condensate, need to be repaired. If there is no danger to personnel or any other compelling reason, the repair should be delayed until there is a convenient shore availability. Fasteners should be checked for sufficient torque as much as possible, since nuts have occasionally been found to be hand-tight. Use a normal-size wrench and try to tighten fasteners using very little torque. If the fastener gives little resistance, turn it until it is snug using mechanic's experience and by evaluating the fastener's condition. If the fastener does not turn, evaluate whether or not to apply greater torque, while being cautious not to strip or break fasteners. In-service leaks at casing horizontal joints or steam chest joints can be pumped (see paragraph 231-8.6.4) if pumping grooves exist. For leaks at pumping groove plugs, see paragraph 231-8.6.2. A leak at a bolt hole can be repaired using a thin (10-mil typical) copper washer under a covered nut that is made to enclose the stud top. Also, bolt hole leaks can be repaired by pumping Copaltite through a pumping stud (Figure 231-8-6), then reinstalling the stud and nut. Allow only one nut and stud to be removed at any one time to maintain joint compression. For steam joints with spiral wound gaskets, see paragraph 231-8.7.7.5. For steam leaks coming through a casting or fabrication, or other major leaks, notify NAVSEA as soon as possible (see paragraph 231-6.1.3.4.h.

231-8.6 CASING JOINT SEALING

231-8.6.1 SURFACES TO BE JOINED. Surfaces to be joined are described in paragraphs 231-8.6.1.1 through 231-8.6.1.5.

231-8.6.1.1 Horizontal Flanges of Casings. The turbine casing is normally of the split configuration. It is split along its horizontal centerline to permit access for inspection or repair of turbine internals.

231-8.6.1.2 Vertical Flanges of Casings. Vertical flanges may also be in the turbine casing. The flanged joint is intended to achieve an economy in production, to use production procedures and facilities best suited to turbine manufacture (cast versus fabricated parts). The majority of these joints, however, are used for production reasons. Once made up and joined, a vertical joint should not be broken during turbine disassembly, as both parts so joined are disassembled as a single part when the horizontal joint is broken and the flanges separated. Some vertical flanges are seal-welded from the inside to ensure tightness.

231-8.6.1.3 Packing Housings of Casings. Packing housings are similarly split horizontally for access (inspection and maintenance) and also have a vertical face for attachment to the turbine casing proper.

231-8.6.1.4 Steam Chests. The chest that receives steam and houses the turbine control valves is properly called the steam chest and not (as is so often the case in correspondence) the nozzle block. The nozzle block is the plate containing the first-stage nozzles through which steam expands from the control valves to the first-stage rotor blading. The steam chest is split horizontally to permit access for inspection or repair of control valves.

231-8.6.1.5 Bearing Housings. The bearing housing may be integral with or separate from the turbine casing and packing housing, but also will be split (cap and base or pedestal).

231-8.6.2 PUMPING GROOVES. Many designs for casing joints consist of mating flanges bolted together without the provision for emergency pumping grooves in either of the two flanges. A simple system of joint grooving, however, is still provided on some turbines, together with arrangements for pressure pumping of these grooves, with an approved form of steam sealing compound. Grooves shall not be pumped, however, except when absolutely necessary in effecting emergency repairs in service. It shall not be used during routine overhaul without NAVSEA approval. A steamtight joint allows no steam leakage to the atmosphere with pumping groove plugs installed. Steam in the pumping grooves is acceptable and does not justify lifting the casing to remake the joint.

231-8.6.2.1 Pumping Groove Plug Removal. Pumping groove plugs should not be removed from a turbine containing steam. If pumping groove plugs leak steam, stop and depressurize the turbine; replace the plugs before trying to pump the groove. Female allen-head plugs may resist removal. After properly grounding the turbine to prevent arcing across the bearing-to-journal clearance, weld the allen wrench to the plug to facilitate removal. Another option is drilling out the plug, which will usually require an oversized pumping groove plug hole with an oversized replacement plug. Check the parts list for oversized plug availability. Protect the pumping grooves from debris as much as possible and use a vacuum often. Remove any cutting oil used to retap threads in the hole using a vacuum, absorbing tools, and appropriate solvents. Completely remove any oil used to chase threads on the plug or in the pumping groove port.

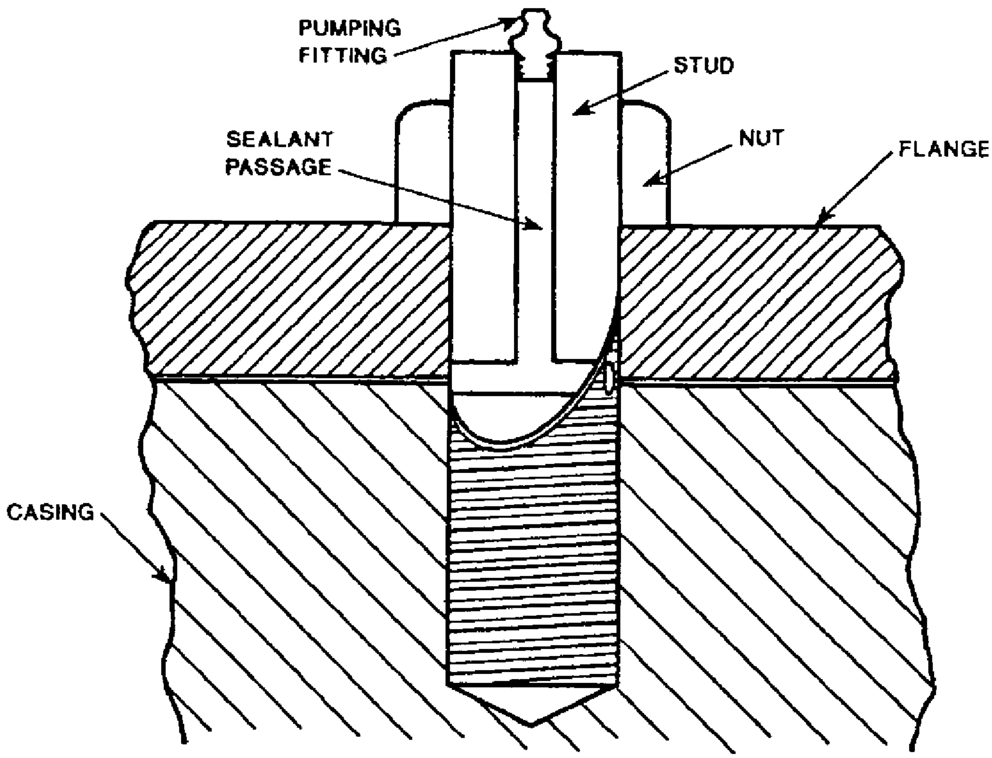


Figure 231-8-6 Pumping Stud

231-8.6.2.2 Pumping Groove Plug Reinstallation. Pumping groove plugs should be replaced with new plugs after removal, if possible, or threads should be chased and the plugs reused. Oversize plugs of proper material may be used (see paragraph 231-8.6.2.1). On installation, pumping groove plug threads should be coated with an approved sealant compound (paragraph 231-8.6.3.2) and torqued to design value, typically approximately 10 foot-pounds, and with proper thread engagement of at least a plug diameter. If steam is not actually leaking from the turbine joint or adjacent bolt holes, do not pump the pumping groove, but rework the plugs. Try this several times before pumping a groove.

231-8.6.3 NORMAL PROCEDURE FOR MAKING UP JOINTS. The procedure for making up joints is described in paragraph 231-8.6.3.1. Joint compound restrictions for nuclear-powered ships are discussed in paragraph 231-8.1.2.

231-8.6.3.1 Steam Chests or Turbine Casings. The steam chest cover-to-head joint or the horizontal-flanged joints of turbine casings shall be cleaned and made up metal to metal, except that one of the flange faces (preferably the upper flange so that dirt does not fall on it) should be coated with a thin coating of an approved joint sealing compound applied at room temperature. Use the compound specified by the turbine manufacturer in the turbine technical manual, if possible. For ships with saturated steam plants, use of the compound shall also comply with NAVSEAINST 9210.36. Procedures for obtaining good surface contact and proper clamping action by bolting are found in paragraphs 231-8.5 through 231-8.5.3.3 and paragraphs 231-8.7 through 231-8.7.7.3. The coating is required for surface imperfections on the joint. Too thick of a coating will cause blowout since the coating is not a sealant.

231-8.6.3.2 Approved Casing Sealing Compounds. Approved sealing compounds (see paragraph 231-8.1.2) are:

a. Bodied linseed oil, also called triple boiled linseed oil, conforming to specification TT-L-201, type II, Linseed Oil, Heat Polymerized, except that viscosity shall be Z-8 or Z-9, and manufactured commercially by many companies. Two sample companies are Wheeler Paint Co., 6343 Penn Avenue, P.O. Box 6566, Pittsburgh, PA 15206 (P/N-OKO-M37) (412-321-1883); and TR3 Company, P.O. Box 427, Columbia, KY, 42728 (502-384-6147).

b. RTV-60, a silicone rubber product of the General Electric Company.

c. SILASTIC E RTV Silicone Rubber Kit (NSN 8030-00-142-9124) or SILASTIC J RTV Silicone Rubber Kit (1 pint base and curing agent), products of the Dow Corning Corp., Government Marketing Services, P.O. Box 0994, Midland, MI 48686, (517) 496-7756, approved for turbines with saturated steam initial conditions. Silastic E is preferred for joints and gunning grooves. Do not use other compounds shown in MIL-S-23586, Sealing Compound, Electrical, Silicone Rubber, Accelerator Required. These other compounds have not passed steam joint testing and have often failed.

d. Copaltite, a product of National Engineering Products Inc., 338 Washington Building, 15th and New York Avenue, N.W. Washington, D.C. 20005. Note that Copaltite will make the joint difficult to disassemble and clean. Use it only in an emergency.

231-8.6.3.3 Packing Housings. Since the housings for gland packing are under low pressure, they are not subject to high-pressure leakage and steam cutting, as are the turbine casings. These housings are made up metal to metal, but a casing-joint sealing compound is used to achieve joint tightness. To prevent steam leakage blue the vertical and horizontal flange in accordance with paragraph 231-8.5.1.1 before assembly.

231-8.6.3.4 Bearing Housings. The turbine bearings are contained in bolted housings which must be oil tight. Common practice is to use some kind of joint sealant, such as Permatex nonhardening or equivalent. Exercise care, however, to prevent such sealants from contacting self-alining surfaces of bearings or contaminating the lube oil system. Use the oil tight joint sealant sparingly in a thin coat stopping 1/4 inch from the edges. See paragraph 231-6.8.1.2 for more information.

231-8.6.3.5 Application Instructions for RTV-60 of Steam Joints for Turbines with Saturated Steam Initial Conditions. RTV-60 silicone rubber forms an effective steam joint compound when cured with a 0.5 percent (by weight) Thermolite T-12 catalyst. This compound is available in a 1-pound packaged form with a premeasured amount of T-12 catalyst from the General Electric Company. The most favorable results are obtained by thinning the compound with an equal volume of axothene or xylene and applying with an ordinary syphon-feed paint spray gun. A roller or a spray can may also be used with an application procedure that has a history of success. When prepared in the procedure given in this paragraph, the kit contains sufficient compound for one turbine casing.

1. Thoroughly clean both joint surfaces with a suitable common solvent (such as axothene, xylene). Mask off the interior steam surfaces of the turbine and any pumping grooves. Verify that masking material is removed before making up the joint. Protect threads and tapped holes during this operation. Prevent dust and foreign material from contaminating the joint.
2. Mix the 1-pound can of RTV-60 with an equal volume of axothene or xylene. This will produce a consistency of approximately that of enamel paint.
3. Add 40 percent of the premeasured packaged catalyst and mix thoroughly. This catalyst concentration (0.2 percent) may be measured by cutting the spout directly below the tip and counting out 54 drops. Complete curing time is in excess of 18 hours.
4. Apply the compound in a thin uniform coat to only one surface. Make no attempt to apply any more compound than is necessary to color the surface.
5. The joint should be snugged up immediately to bring both surfaces together because the compound will begin to set in approximately 3 hours at room temperature. Final tightening should be accomplished as soon as possible since any hardened compound will hold the joint apart.

231-8.6.3.5.1 RTV-60 with expired shelf life may be used if ASTM Standard Test Method D2240 type A shore durometer hardness exceeds 55.

231-8.6.3.5.2 For turbines with superheated steam initial conditions, the same application instructions apply, except uncatalyzed or catalyzed RTV-60 may be used. Although the cure time is not critical for the uncatalyzed, close the joint as soon as the surface of the joint is properly coated so that foreign material will not contaminate the compound.

231-8.6.3.5.3 Dow Corning SILASTIC E or J Silicone Rubber kits older than their 6-month shelf life may be used if ASTM Standard Test Method D2240 type A shore durometer hardness exceeds 30 for SILASTIC E or D2240 type A2 shore durometer hardness exceeds 50 for SILASTIC J Silicone Rubber. Mix compound as directed on containers, except that a vacuum chamber need not be used on the compound for this application. Joint surfaces (or pumping groove opening surfaces) shall be free of oil and should be as close to 77°F as possible. Batch-test the compound by applying part of the batch using the same method to sheet mylar or glass or similar surface. After the joint surfaces have passed a blue check for contact and have been cleaned, apply a uniform 1- to 2-mil-thick layer of compound to the top joint surface. This should be no thicker than needed to color the surface. A print roller may be used. Final tightening should be accomplished as soon as possible since hardened compound will hold the joint apart. After the joint is made up and cured, remove the batch-test sample and verify that it remains in a plastic-like sheet.

231-8.6.4 EMERGENCY PROCEDURE FOR PUMPING FLANGE GROOVES. The procedure for pumping flange grooves is discussed in paragraphs 231-8.6.4.1 through 231-8.6.4.9.

231-8.6.4.1 Purpose of Grooves. Some turbine casings are provided with grooving in one flange so that flange steam leaks can be temporarily stopped by pressure pumping a sealant into the grooves. If a leak can be isolated to the vicinity of one of the several grooves in a joint, then just that one groove (not all grooves) needs to be pumped. The same logic holds for just 2 grooves.

231-8.6.4.2 Proper Sealant Compound to Use. Pumping grooves may be pumped with Copaltite of proper consistency and viscosity to completely fill the grooves. For proper consistency do not use Copaltite that is more than 1 year beyond date of manufacture shown on container. Order Copaltite type C of MIL-S-15204, Sealing

Compound, Joint and Thread, High Temperature, with a viscosity of 250,000 centipoise (minimum) to 300,000 centipoise (maximum) specifically for pumping grooves. Suitable compound is available from National Engineering Products, Inc. (paragraph 231-8.6.3.2). Pumping grooves in turbines with saturated steam initial conditions may be pumped with SILASTIC E or SILASTIC J (paragraph 231-8.6.3.2). SILASTIC J is preferred. Consistency is correct as mixed by container directions if it is within the 6-month shelf life. If outside the shelf life, test the compound in accordance with paragraph 231-8.6.3.5.3. Working time for SILASTIC E is 2 hours after the curing agent is added; working time for SILASTIC J is 2.5 hours. After mixing in a container three times too large to allow for expansion, place in a vacuum chamber at 28 inches mercury for 5 minutes to remove trapped air.

231-8.6.4.3 Preuse Test of Copaltite. Before using, check the Copaltite viscosity by coating a 1x3-inch steel plate with a 1/8-inch thickness of compound. The test plate surface should be free of rust, scale, and organic matter. The coated plate should be suspended vertically at room temperature 75°F for 1 hour. The compound should neither sag nor flow during the 1-hour test. Viscosity cannot be increased under field conditions. The compound is too stiff if it will not spread easily and adhere to the plate. The viscosity may be decreased by adding small amounts of methanol. Do not use the compound if it has jelled and will not mix with methanol.

231-8.6.4.4 Preparation for Pumping. The pumping may be done using either a pressure pumping outfit or a grease gun capable of 5,000 psi discharge pressure. The standard grease gun for SILASTIC compounds should be similar to model 1142, series B manufactured by Lincoln Engineering Company, St. Louis, MO. Equipment shall be thoroughly cleaned before use. Thoroughly clean the grooves, and make sure that the flange temperature is less than 150°F to ensure proper flow of the Copaltite. The flange should be at ambient room temperature before using a SILASTIC compound. Using compressed air attached to an end plug of the groove, blow through the adjacent hole with the remaining plugs in place. Plug the air exit hole, and blow through the next hole. Continue this process, clearing in sequence each section of the groove from the end hole.

231-8.6.4.5 Procedure for Pumping. The pumping of Copaltite should be done with a vacuum on the groove to minimize voids resulting from air entrapment. Attach the pumping gun to the end plug. Thread a vacuum tap into the adjacent hole with all other plugs in place. Using a vacuum source (capable of drawing a vacuum of at least 5 inches Hg and at least 6 inches of clear tubing), draw a vacuum on the groove while pumping until compound shows at the vacuum connection. Special fittings are required on the tubing to permit threading into the plug hole and to the vacuum source. Remove the gun and plug the hole. Remove the vacuum tap and attach the pumping gun. Thread the vacuum tap into the next hole, and again pump while drawing a vacuum until compound runs out the vacuum tap. Each time the vacuum tap is relocated, the tube should be rodded out and rinsed with alcohol. Proceed in this manner until all holes are pumped and plugged. On the last hole the pumping is done with no vacuum to force compound into any dead ends. Pump this last hole slowly until a pumping effort of about twice that required on previous holes is exerted. Accomplish the pumping sequence without interruption to prevent the compound from setting before the pumping has been completed. Pump SILASTIC compounds without a vacuum.

231-8.6.4.6 Postpumping Curing. After pumping and thoroughly resealing all plugs, the Copaltite shall be cured to provide a rubbery, pressure-retaining seal. Heat the flange to 200°F to 250°F for 24 hours. Curing at a temperature above 250°F results in a porous seal because of bubble formation. Steam or strip heaters may be used to apply heat. SILASTIC compounds cure properly at ambient room temperature in 24 hours; heat is not required.

231-8.6.4.7 Temporary Fix. Pumping turbine grooves is a temporary fix. The pumped turbine joint should be cleaned of pumping compound and remade to be steam tight the next time that joint is opened for other reasons, or at next ship availability if that joint is found to be leaking in spite of the compound or if an unapproved compound has been used.

231-8.6.4.8 Use of Pumping Studs. Some turbine casings leak where there is no pumping groove (for example, the exhaust casing). Use of a pumping stud (Figure 231-8-6) is acceptable. Remove the pumping stud and torque the correct fasteners to design torque before pressurizing the turbine. Allow only one nut and stud to be removed at any one time to maintain joint compression. Another acceptable and effective fix for a leak into the exhaust casing has been triple-boiled linseed oil drawn into the joint under vacuum.

231-8.6.4.9 Report Required. Submit to NAVSEA a report on each instance of groove pumping. The report should detail the location and extent of leakage, any difficulties encountered during the pumping, and the results of the pumping. Reports should be made by naval message or letter to NAVSEA code 03Z23 within 30 days of the repair.

231-8.7 MAKING UP TURBINE CASING BOLTING

231-8.7.1 TIGHTENING. Proper tightening of turbine horizontal casing joint and valve chest cover bolting is essential to obtain the clamping force required for satisfactory performance of metal-to-metal steam joints. The clamping force exerted by a bolt is a function of tensile preload, or bolt stretch, resulting from tightening and can be determined by measuring:

a. The bolt length before and after tightening

b. The torque applied to the nut

c. The advancement of the nut on the bolt thread

231-8.7.2 HELICOILS. Helicoils are an acceptable repair method when properly installed for hole sizes up to 2.0 inches. Local waivers are allowed. Verify casing integrity and proper installation. Record helicoil installation for configuration purposes if required.

231-8.7.3 MATERIALS. Two types of materials are generally used for turbine-joint bolting: alloy steel and carbon steel. Alloy-steel bolting is used where temperature or stress limitations for carbon steel are exceeded. Present design practice specifies alloy steel (MIL-S-1222, type 1, symbol B-16, Bolts - Studs, Nuts, and Bars, Round; Steel) for all applications in which the maximum expected temperatures do not exceed 900°F. For all applications designed to operate above this temperature, the bolting is made of MIL-S-861, Steel Bars, Corrosion Resisting, Naval Steam Turbine Parts Use, class 422 (corrosion-resistant chrome) material. Whenever turbine-joint bolting is replaced, follow this design practice unless specifically approved by NAVSEA or bolting is in accordance with OEM drawing. Experience has shown that many steam leaks have resulted from repair activities using incorrect bolting materials. Design standards require the alloy material to be stretched 0.0015 inch for each inch of free (effective) length of bolt. The carbon-steel bolting is stretched 0.001 inch for each inch of free length. These correspond nominally to 45,000- and 30,000-psi stress, respectively. In addition, some units have bolting stretched to 60,000- to 90,000-psi stress. The turbine technical manuals provide instructions for tightening to these stresses.

231-8.7.4 LENGTH. The effective length of the bolt or stud can be determined as follows:

a. Bolts. The distance from the head of the bolt to the nearest engaging thread plus half the bolt diameter

b. One-Nut Studs. The distance from the nut face to the nearest engaging thread plus one stud diameter

c. Two-Nut Studs. The distance between the nut faces plus one stud diameter.

231-8.7.5 REPLACEMENT. Turbine-casing horizontal joint bolting need not be replaced each time the joint is broken and with proper handling should last the life of the turbine. It is also unnecessary to renew the valve chest bolting each time the joint is broken. Good practice, however, requires that the bolting be replaced as part of a routine valve overhaul for 950°F steam service. This bolting shall be made from MIL-S-861, class 422 (paragraph 231-8.7.3).

231-8.7.6 INSTRUCTIONS. Detailed instructions for making up turbine-casing bolting are usually included in the manufacturer's instruction book and, if provided, should be followed. In the absence of such instructions, paragraphs 231-8.7.6.1 and 231-8.7.6.3 are offered as guidance. Experience has shown that when new bolting is installed, particularly steam chest bolting exposed to temperatures above 900°F, the bolting should be rechecked and retightened after initial dockside steaming and again after sea trials following the overhaul. All checks and retightening are to be done with bolting at or near room temperature. This procedure will reduce the potential for steam to leak at high-temperature steam-chest joints after the joints have been remade with new bolting.

231-8.7.6.1 Preparatory Work. Clean all joint studs, nuts, and washers thoroughly to remove any previously used thread compound or foreign material. Inspect threads for damage, such as galling, wire edges, and loose slivers. Replace any studs that have damaged threads. The studs may come out when nuts are being removed, but this is acceptable. Remove nuts, repair or replace the studs and nuts, and then reinstall the studs with the proper torque (see paragraph 231-8.7.6.1.2). A stud may spin in its hole. Establish the correct thread fit or bottom the stud correctly (see paragraph 231-8.7.6.1.2). Inspect nut and washer faces for galling, and remove raised metal by filing. Check all fasteners for symbol identification. If no symbol is visible, do not use the fastener unless the material has been verified by a local materials lab. Check nuts to ensure that they can be turned by hand through the full thread engagement on their respective studs. Failure to turn freely by hand may indicate that the stud or nut threads have been stretched. In such cases, replace the defective part. All replacement bolting shall have a class 2A or 3A thread fit with casing tapped holes. Check installed studs with closed or cap nuts for a minimum of 1/8-inch clearance between the end of the stud and the nut, to prevent bottoming during tightening. Thicker washers may be used to provide additional clearance.

231-8.7.6.1.1 Before final installation, coat stud threads and nut faces with antiseize compound MIL-A-907, Antiseize Thread Compound, High Temperature. Follow the instructions in paragraph 231-8.6.3 on applying joint-sealing compound to flange faces.

231-8.7.6.1.2 Before installing bottoming studs, measure the depth of the casting remaining under the subject holes. Then calculate the minimum depth of the casting required to support torquing of the bottoming studs without cracking. All bottoming studs shall be wrenched until they are tightly bottomed using a strap or stud wrench, but not a pipe wrench.

231-8.7.6.2 Bolt-Tightening Sequence. Tighten the horizontal-casing joint bolts and the steam chest bolts from the center of the casing toward either end, alternating from one side of the turbine to the other (Figure 231-8-7). Tightening in this manner will force any excess joint compound to flow from the joint faces and minimize the effect of any distortion in the joint flanges.

231-8.7.6.3 Measuring to Determine Bolt Stress. Tighten all nuts with a 200-pound pull on a wrench with a handle length of 1 foot (200 lb/ft torque for each inch of bolt diameter). This will establish the reference point, zero stretch, from which the bolt stretch will be measured and bolt stress established. Take zero-stretch micrometer measurements for all bolts that can be measured, and record them. Obtain the number of required nut turns from the appropriate chart for the bolting material used (Figure 231-8-8 through Figure 231-8-10 and Table 231-8-1). Chalk mark reference points on the nuts and adjacent flange. Turning the nuts until the marks coincide should produce the required bolt stress. Verify proper pretorque of each stud after tightening adjacent stud to chalk mark. Where possible, remeasure bolts after tightening under the cold condition of zero measurement to verify the correct stretch. The permissible tolerance for bolt stretch is + 10 percent or 0.002 inch, whichever is less. Correct errors greater than the tolerance. Bolts that cannot be measured and are below the range of the nut-turn chart are to be stretched using the torque given in Table 231-8-1, Table 231-8-2, and Table 231-8-3, as appropriate.

231-8.7.7 TIGHTENING METHODS

231-8.7.7.1 Cold Tightening. Cold-tighten all bolts and nuts up to 2 inches in diameter with extension wrenches only. If space limitations prohibit using extension wrenches, the order of preference is hydraulic wrenches followed by heating rods and slugging wrenches. Using other than extension wrenches of the proper size may cause threads to gall or spot faces to score.

231-8.7.7.2 Measuring Bolt Stretch. There are three techniques for measuring the correct bolt or stud stretch. When a choice exists, the preferred order of accuracy is as follows:

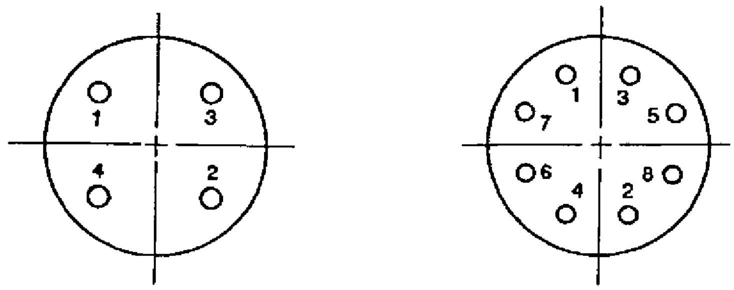
a. Directly measuring stretch

b. Measuring flats or degrees of rotation

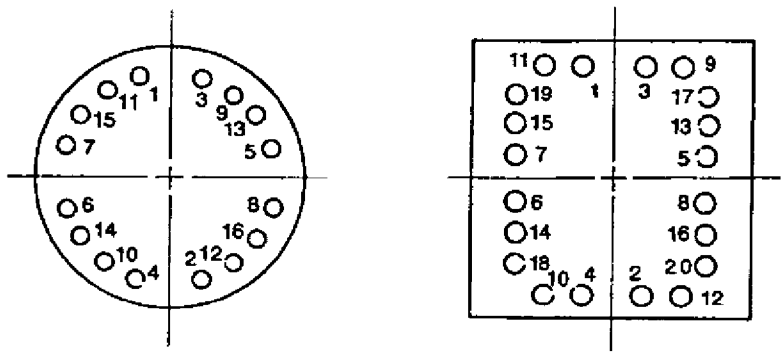
c. Measuring torque with a torque wrench

231-8.7.7.2.1 The only positive way to measure the amount of bolt stretch is with a micrometer. The degree method should always be verified by stretch for sizes 2 inches and greater. The torque method should be verified by stretch on the first stud torque for each size 2 inches and greater. Bear in mind that there are drawbacks to using torque as an indication of bolt stretch. Factors such as thread finish, nut face, washer finish, and thread compound can significantly change the coefficient of friction and therefore the bolt stress resulting from a given torque. In no case should the nut advancement, when torque is measured for bolt sizes included in the appropriate nut turn chart (Table 231-8-1 and Figure 231-8-9 and Figure 231-8-10), be less than the amount specified by the chart.

231-8.7.7.3 Hot Tightening. Bolting 2 inches in diameter or larger and designed with a centerline heating hole shall be tightened hot. Heating considerably reduces contact pressure at the various sliding surfaces and eliminates the possibility of galling the threads, nut faces, washers, or spot faces with the high forces produced by sledging or using long wrench extensions. Bolts tightened with heat should also be heated before loosening during disassembly. Proper use of bolt heaters will eliminate the need for the wrench extensions or sledging that can damage threads. Either an electric heating element or a specially designed gas torch may be used to heat the bolt.



TYPICAL FLANGE JOINTS



TYPICAL ONE-PIECE CASING

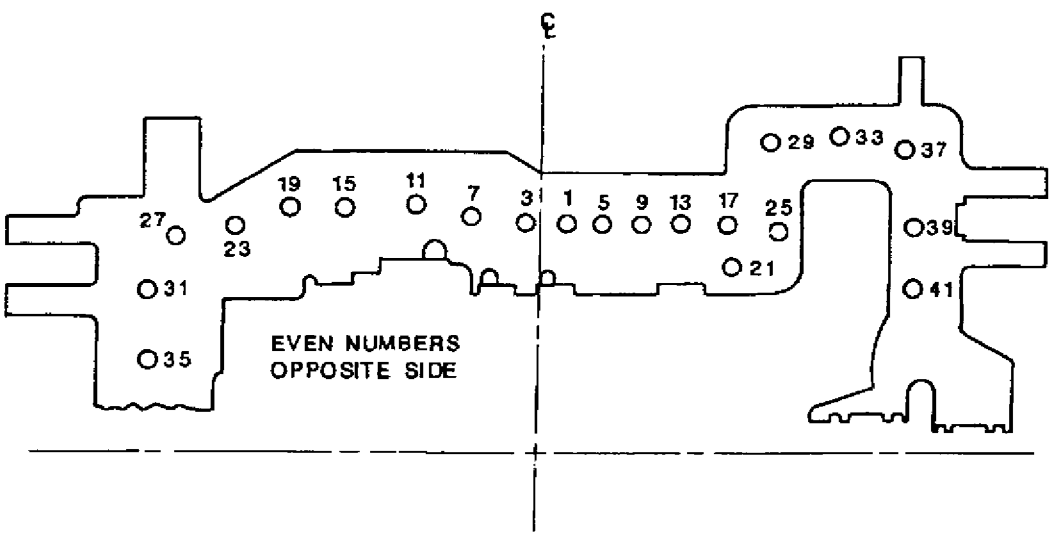


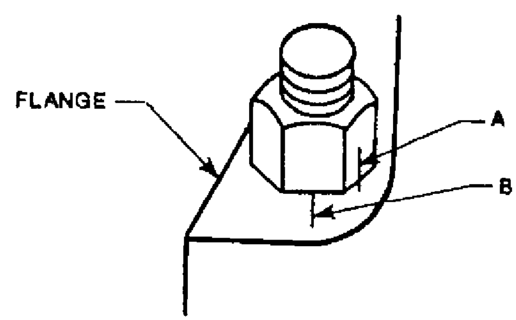
Figure 231-8-7 Markings for Sequential Tightening

ONE– AND TWO–NUT STUDS – 45,000–PSI STRESS

OVERALL LENGTH OF STUDS	4–4 7/8	5–5 7/8	6–6 7/8	7–7 7/8	8–8 7/8	9–9 7/8	10–10 7/8	11–11 7/8	12–12 7/8	13–13 7/8
NO. OF FLATS	FL	FL	FL	FL	FL	FL	FL	FL	FL	FL
1, 1–1/8, 1–1/4, 1–3/8	3/8	1/2	5/8	3/4	1	1	1–1/8	1–1/8	1–1/4	1–1/4
1–1/2, 1–5/8, 1–3/4, 1–7/8	3/8	3/8	1/2	3/4	7/8	1	1	1–1/8	1–1/8	1–1/4
STUD DIA 8–PITCH										

BOLT (HEX. OR SOCKET HEAD) – 45,000–PSI STRESS

LENGTH UNDER HEAD		1–1 7/8		2–2 7/8		3–3 7/8		4–4 7/8		5–5 7/8		6–6 7/8		7–7 7/8	
NO. OF FLATS DEGREES		FL	DEG	FL	DEG	FL	DEG	FL	DEG	FL	DEG	FL	DEG	FL	DEG
BOLT DIA.	3/4, 7/8	1/4	12°	1/4	17°	3/8	26°	1/2	34°	3/4	43°	7/8	52°	1	63°
NC & 8–PITCH	1, 1/8, 1–1/4, 1–1/2, 1–3/4	1/8	8°	1/4	12°	3/8	19°	1/2	26°	5/8	34°	3/4	41°	7/8	51°



SUGGESTED MARKING OF NUTS AND FLANGE
FOR REFERENCE WHEN APPLYING FLATS DATA

Figure 231-8-8 Torque and Flats Data, Tightening, and Bolts

STRESS 45,000 PSI

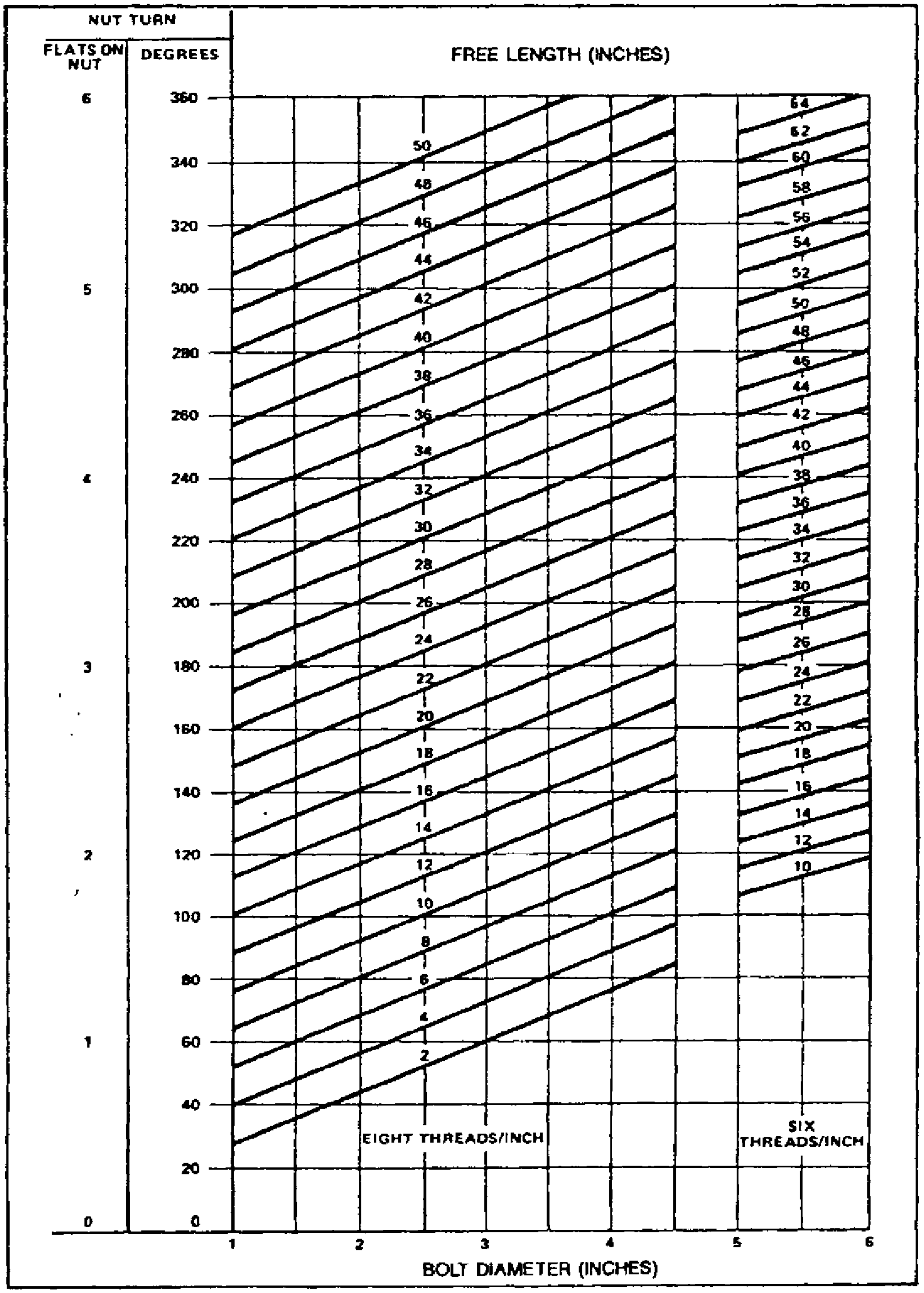
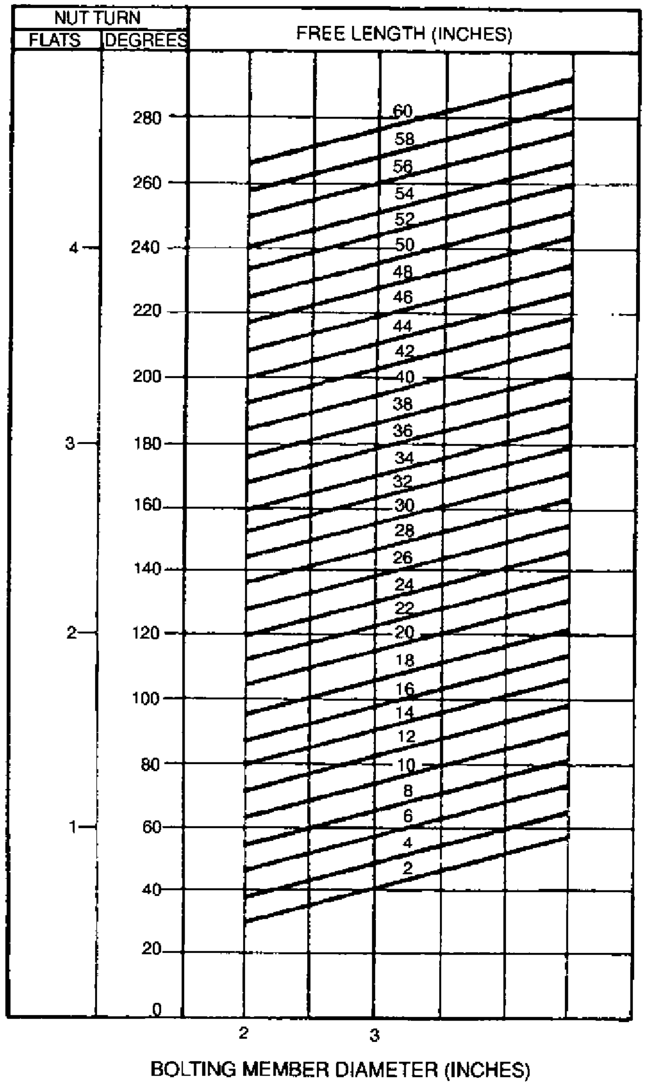


Figure 231-8-9 Alloy-Steel Bolting Nut-Turn Chart

STRESS 30,000 PSI



NOTE: CURVES ARE FOR EIGHT-THREAD BOLTING MEMBERS ONLY AND ARE BASED ON 4 X FREE LENGTH PLUS 11 X DIAMETER.

SAMPLE CALCULATION

(A) THREAD DIAMETER OF STUD.. 2 INCHES

(B) THICKNESS OF FLANGE FROM JOINT FACE TO SPOT FACE = 6 INCHES PLUS ONE STUD DIAMETER, EQUALS FREE LENGTH OF................. 8 INCHES

(C) HEX FLATS TO BE TURNED (FROM CHART ABOVE)............................1 (OR 60 DEGREES) APPROXIMATELY

Figure 231-8-10 Carbon Steel Bolting Nut Turn Chart

Table 231-8-1. BOLTING STRETCH OR DEGREES NOT FOR HYDROTEST*

Standard 30,000-psi Stress Eight-Pitch Threads					
Effective Fastener Length (EFL)	**Degrees +3 -0**	**Stretch +0.002 -0.000**	**Effective Fastener Length**	**Degrees +3 -0**	**Stretch +0.002 -0.000**

Table 231-8-1. BOLTING STRETCH OR DEGREES NOT FOR HYDROTEST* - Continued

Standard 30,000-psi Stress Eight-Pitch Threads					
1.00-1.38	4	0.001	10.55-11.38	40	0.011
1.50-2.38	8	0.002	11.55-12.38	44	0.012
2.50-3.38	11	0.003	12.50-13.38	48	0.013
3.50-4.38	15	0.004	13.50-14.38	50	0.014
4.50-5.38	19	0.005	14.50-15.38	54	0.015
5.50-6.38	21	0.006	15.50-16.38	58	0.016
6.50-7.38	25	0.007	16.50-17.38	61	0.017
7.50-8.38	29	0.008	17.50-18.38	65	0.018
8.50-9.38	33	0.009	18.50-19.38	69	0.019
9.50-10.38	36	0.010	19.50-20.38	73	0.020

NOTE:Stretch values indicated shall be used only with an extensionmeter measuring gage. When a depth micrometer special and or standard is used, multiply the stretch values indicated by 1.25 inches to obtain the stretch value required when using these instruments.

Table 231-8-2. ALLOY NUTS

Diameter (Inch)	Threads (Inch)	Torque (In-lb)
1/2	13	540
5/8	11	1,080
3/4	10	1,800
7/8	9	2,900
1	8	4,410
1-1/8	8	6,400
1-1/4	8	9,000
1-3/8	8	12,250
1-1/2	8	14,400
1-5/8	8	19,800
1-3/4	8	27,000
1-7/8	8	36,000
2	8	39,600

Table 231-8-3. CARBON-STEEL NUTS

Diameter (Inch)	Threads (Inch)	Torque (In-lb)
1/2	13	360
5/8	11	720
3/4	10	1,200
7/8	9	1,930
1	8	2,940
1-1/8	8	4,260
1-1/4	8	6,000
1-3/8	8	8,160

Table 231-8-3. CARBON-STEEL NUTS - Continued

Diameter (Inch)	Threads (Inch)	Torque (In-lb)
1-1/2	8	9,600
1-5/8	8	13,200
1-3/4	8	18,000
1-7/8	8	24,000
2	8	26,000

231-8.7.7.4 Torquing. Fasteners requiring torquing should normally be torqued to the drawing requirements. Larger fasteners, however, may be tightened by other methods as appropriate. Also, some fasteners cannot be reached with a torque wrench because of interference. In this case, measure the number of flats rotated on a properly torqued, accessible fastener, and tighten the inaccessible fastener the same number of flats.

231-8.7.7.5 Spiral-Wound Gaskets. For joints with spiral-wound (flexatalic) gaskets, use the following tightening procedure:

1. Verify that the joint flange surfaces, grooves, and raised faces are according to the drawing.
2. Use a correct, new spiral-wound gasket. Position it correctly.

CAUTION

For initial torquing, tighten gasket to no more than 30 percent of the required bolt stress. Overtorquing could cause gasket damage that cannot be corrected by further tightening.

3. Torque to no more than 30 percent of the required bolt stress.
4. Develop the required bolt stress in a minimum of four steps. Bring flange faces together parallel with the aid of feeler gages. Tighten diametrically opposite fasteners.
5. The final torquing should be done in a clockwise bolt-to-bolt sequence and in 10 percent increments to ensure that all bolts are evenly stressed.

231-8.8 BOLT HEATERS

231-8.8.1 ELECTRIC BOLT HEATERS. Electric bolt heaters are bayonet type, consisting of resistance wire wound on a ceramic core and enclosed in a stainless steel sheath. They can usually be operated on 110-120 volts ac in circuits employing standard protective devices. Heaters are to be grounded as required for shipboard use. To avoid unnecessary delays, a minimum of four heaters of sufficient length to uniformly heat the effective length of the studs should be available for the heating operation.

231-8.8.2 APPLICATION AND PRECAUTIONS. After preliminary tightening, measuring, and marking of all studs and nuts for the required advancement, insert two heaters, one each in the first stud to be tightened on either side of the casing. Energize heaters and tighten the nut as the bolt expands.

NOTE

On two-nut studs, one nut should be marked for position and held stationary while the other nut is advanced (Figure 231-8-8).

231-8.8.2.1 Continue heating until the nut can be advanced the required amount; however, if the desired tightening is not obtained after heating 7 minutes for each inch of bolt diameter, it is advisable to investigate the cause to avoid unnecessary burning out of the heater.

WARNING

To avoid danger of electric shock and burns, always de-energize heaters before handling.

231-8.8.2.2 When tightening is completed, de-energize and remove heaters.

231-8.8.2.3 Insert heaters in the four adjacent studs (next four studs in the tightening sequence). Continue tightening in this manner until all studs requiring heating have been tightened.

231-8.8.2.4 After studs have cooled to room temperature, measure stretch and correct if beyond tolerance.

231-8.8.3 GAS-TYPE BOLT HEATERS. Gas-type bolt heaters of several different designs are furnished by the turbine manufacturer when gas is the desired heating medium for stretching casing bolting. The various designs may use an oxyacetylene torch, propane torch, or combinations of these, with compressed air. Compressed air is heated by the flame of the oxyacetylene torch. Hot gases heat the stud and exhaust upward outside the heater pipe. When the heater is used on a through stud, plug one end of the stud. When gas-type bolt heaters are provided, follow the manufacturers' instructions for their use. A special tip can be made for an oxyacetylene torch for heating bolts in the absence of other heaters, and when flame heating is not specifically prohibited by the manufacturer (Figure 231-8-11). This heater can be used only on studs that are drilled full length and unobstructed by measuring plugs. Never use it on studs threaded into a blind hole in the casing flange. Gas-type heaters are not permitted on submarines.

231-8.8.4 APPLICATION AND PRECAUTIONS. The procedure for using gas-type bolt heaters is similar to that described for electric heaters, except that bolts are heated and tightened one at a time.

231-8.8.4.1 When using a gas-type bolt heater, assemble a heating chamber, which includes a 1/4-inch heater pipe that is 75 percent plus 15 percent of the overall stud length (Figure 231-8-12). If the heat chamber cannot be assembled because of physical restrictions, and heat has to be generated directly in the stud hole, move the torch continuously up and down in the stud hole. This distributes the heat evenly along the fastener and prevents damage from localized overheating. Heat bolts approximately 2 minutes before tightening.

231-8.9 REMOVING OF BLADING FOR EMERGENCY OPERATION

231-8.9.1 CIRCUMSTANCES REQUIRING BLADE REMOVAL. Casualties have occurred where certain rows of rotor or cylinder blades were damaged to such an extent that the turbine could not operate with the damaged blading in place. The causes of such casualties include cracked or torn blades or shrouds, and missing blades or shroud segments. Normally, in such cases, spare blades are installed or, if available and feasible, a replacement rotor shall be installed. If, however, there is insufficient time to permit reblading, if spare blades or other replacement material are unavailable, or if a design deficiency is suspected and time is required to analyze the failure and provide redesigned material, cut off the damaged blades so that operation may continue until the permanent correction can be accomplished. For the particular corrective action, all decisions regarding removal of blades shall be referred to NAVSEA.

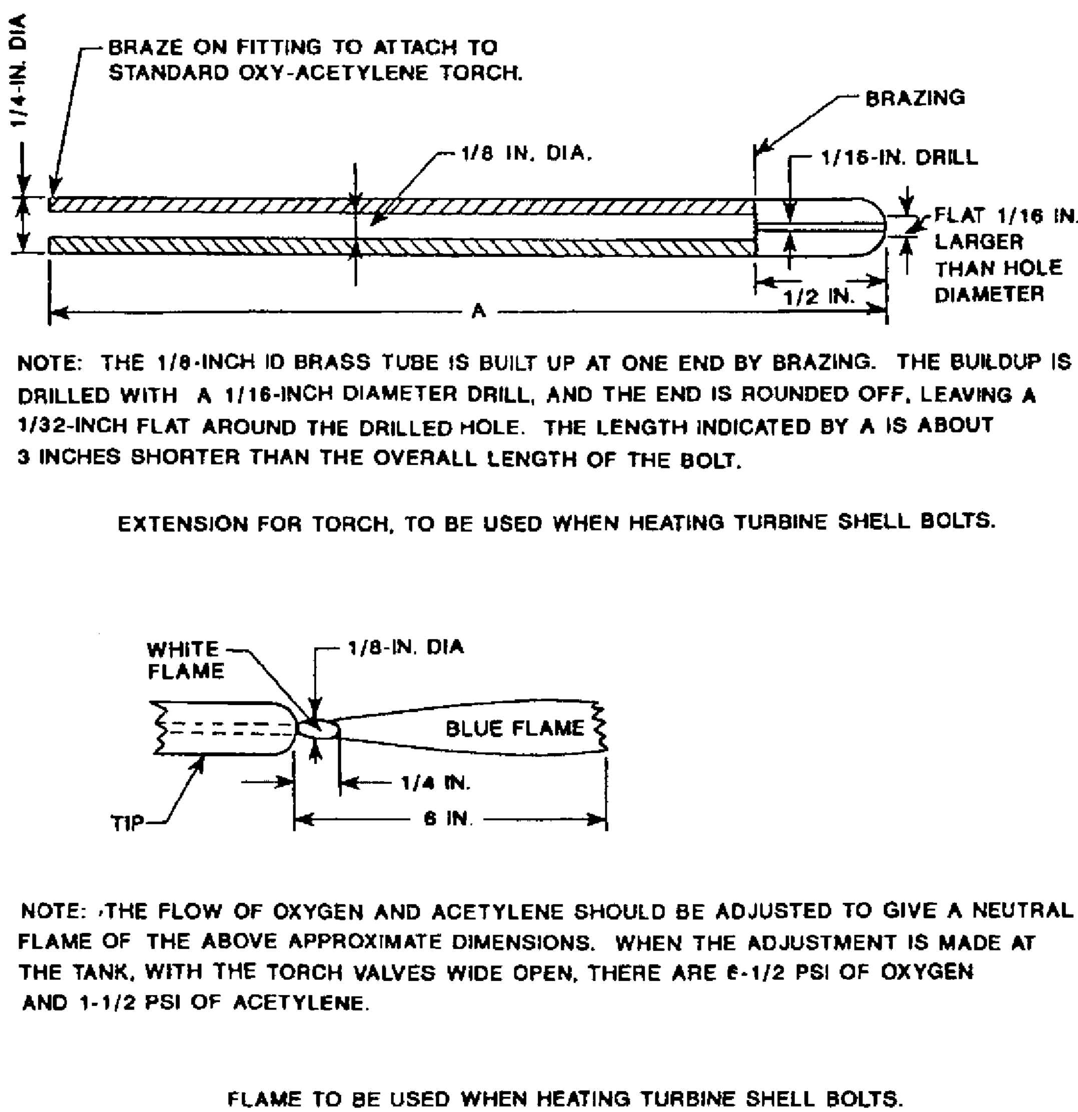


Figure 231-8-11 Gas Bolt Heater

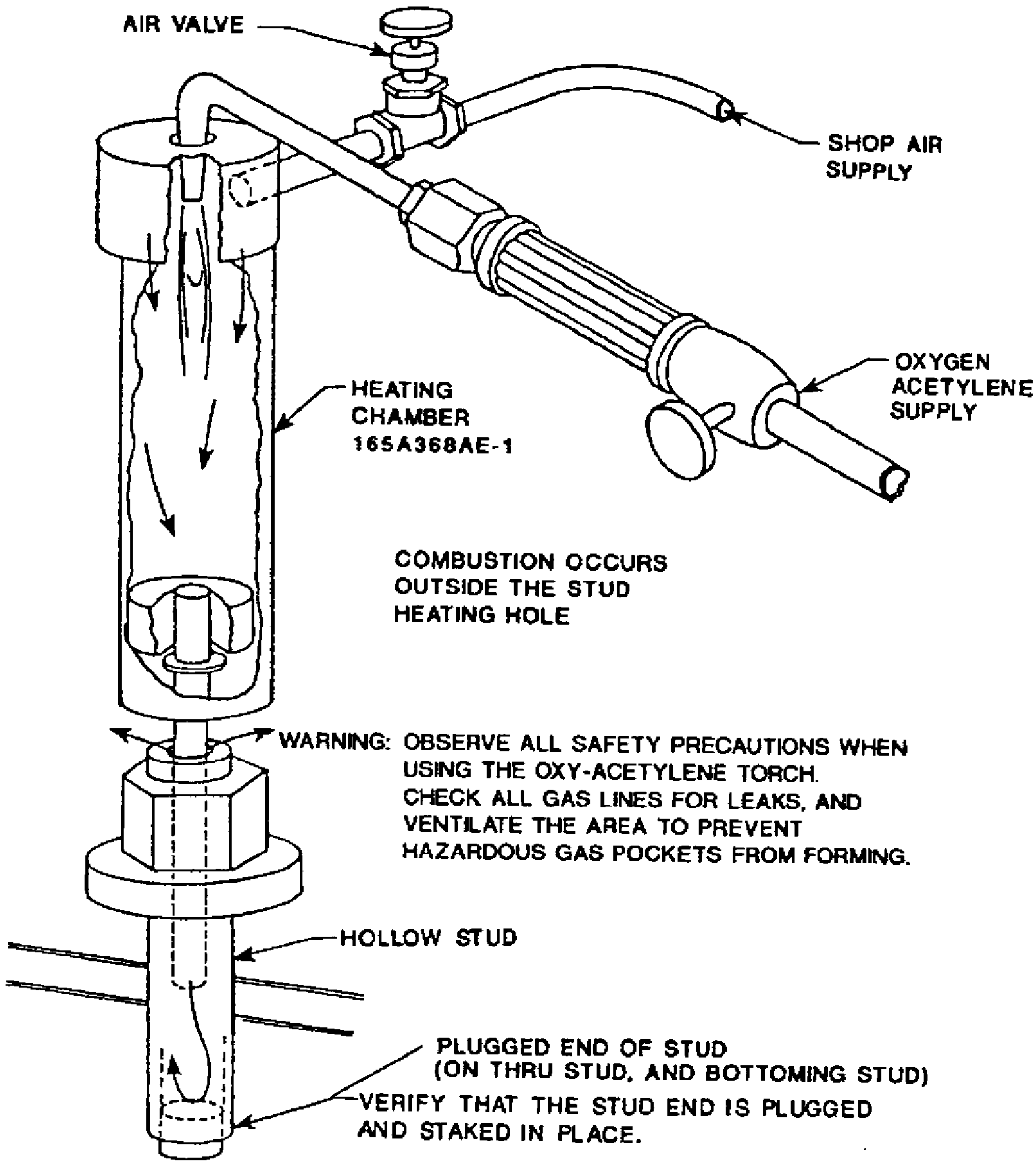


Figure 231-8-12 Heating Chamber Setup

231-8.9.2 WHICH BLADES REQUIRE REMOVAL. In determining whether to remove individual blades, a shroud group, or the entire row, consider the probable cause of the damage and the reliability of the remaining blades.

231-8.9.2.1 If it is apparent that the damage was caused by handling, the passage of foreign material, or other circumstances unlikely to recur, and a single blade or several adjacent blades have been damaged, these may be removed with a clean cut close to the blade platform. Remove a shroud grouping entirely if it does not consist of at least four blades. To restore approximate rotor balance, remove an equal number of blades from a section diametrically opposite on the rotor in the same row. In special situations, removing blades at three positions is acceptable if a vector plot of removed blades shows a canceling of forces or if balancing is to be accomplished.

231-8.9.2.2 Where blade vibration is the known cause of failure and prescribed action is being taken to eliminate the vibration (that is, installation of a redesigned diaphragm), a limited number of blades may be removed.

231-8.9.2.3 In any case where the cause of failure cannot be otherwise identified, suspect blade vibration. When blade vibration is suspected - and in particular when the failures have occurred in the tenon areas where, because of the assembly of the shroud band, the initiation of other cracks cannot be detected - the reliability of the entire row of blades is in question. In such cases remove all the blades in that row.

231-8.9.2.4 Historically, blades have been thrown off the wheel and sometimes through the casing only by an uncontrolled overspeed condition. Blade vibration (resonance) failures have resulted in popped tenons, cracked blades, cracked or lifted shrouds, but not historically in thrown blades.

231-8.10 INSTRUCTIONS ON HOW TO REMOVE BLADES IN PLACE

231-8.10.1 ROTOR BLADE REMOVAL. To protect wheel rims or blade grooves against corrosion and erosion, the root sections of the blades removed shall remain in place wherever practical until new blades are installed. Refer to NAVSEA for a decision if leaving roots in place is impractical because of root failure. Cut off blades at the bottom of the steam passage or at the vane portion of the blade, at uniform height, to maintain rotor balance. Assorted rotary files driven by a small high-speed air turbine have been found useful for cutting off individual turbine blades. In an emergency and when blades of an entire row are to be removed, they may be machined off with a cutting tool supported from the horizontal joint while the rotor turns in its bearings. An axial cut just above the blade platform is required. Stop the cut before the blades are free so that the blade groups can be broken off to prevent them from falling into the casing.

231-8.10.2 DIAPHRAGM OR STATIONARY BLADE DISPOSITION. In the case of an impulse (diaphragm-type) turbine, all diaphragms should remain in place when any row of rotating blades is removed. This will keep the pressure drop across all diaphragms close to the design drop and protect the casing diaphragm grooves and seating surfaces from corrosion and erosion.

231-8.10.2.1 In the case of a reaction turbine, all stationary blading should remain in place if at any time it becomes necessary to remove any rotating blading. With removal of any stationary blading because of damage, all reliable rotating blading should remain in place.

231-8.11 REPAIRING MINOR DAMAGE TO BLADES

231-8.11.1 DENTS AND TEARS. In many instances turbine blades are found dented or torn by the passage of foreign material. In a majority of cases, the extent of damage does not warrant renewal of these blades, but repair action is recommended. Work the dented blades back to their original form, and remove all tears. A pencil grinder or rotary file can be used to dress torn edges, and radius the local area so that no sharp corners remain.

231-8.11.2 EROSION AND CORROSION PITTING. Erosion and corrosion pitting of blades usually results in rough, ragged, or lacy edges, which look very bad but, except in special cases where blade vibration is a factor, have not significantly weakened the blade or reduced the overall machine efficiency. When edges are excessively thin and perforated so that edge flutter may occur and cause bits to break away, such edges should be ground back sufficiently to obtain heavier, smooth edges. When thin edges are cut back, the edge radius may be restored with a notched scraper. If doubt exists about the seriousness of the condition, NAVSEA may be contacted to determine whether there is any history of blade failure in the stage in question. Under no circumstances should any significant grinding operation be undertaken solely to improve appearance.

231-8.11.3 SHROUDING AND TENONS. Stationary blade shrouding and tenons are sometimes damaged by rubbing or improper rotor handling and, in rare cases, may fail because of vibration. If the blading itself is sound, the shroud bands may be replaced, while broken tenons may be repaired by plug welding. The damaged bands can be removed by drilling into the tenons just enough to free the band. A template is then made from the remaining tenon stubs, and a new band section is fitted and attached by plug welding. Do not plug weld or repeen tenons on rotor blades.

231-8.12 REPLACING BLADING

231-8.12.1 GENERAL. All turbine rotor reblading shall be accomplished by qualified bladers from the turbine manufacturer’s plant or under the direct supervision of a technical representative who is currently employed by the original turbine manufacturer. Basic root designs of blading are described in paragraphs 231-8.12 through 231-8.12.3.

231-8.12.2 REPLACING CIRCUMFERENTIAL-ENTRY-TYPE ROTOR BLADING WITH INTERFERENCE ROOTS AND RIVETED TENONS. Each design requires special consideration when removing or installing blades, and applicable technical manuals and drawings must be consulted to obtain root dimensions, details of the locking piece, and any special instructions pertaining to the design. When an entire row of blading is to be replaced, remove the rotor from the turbine and place it in a lathe, if possible. When necessary, reblading may be accomplished in place by turning the rotor in its bearings. When machining is involved, however, it is preferable that separate bearings and pedestals be provided so that the rotor may be lifted out of the casing. This avoids damage to blading and seals and plugging of drains by debris.

231-8.12.3 REMOVING OLD BLADES. Old blades are removed by machining or driving. These methods are described in paragraphs 231-8.12.3.1 and 231-8.12.3.2.

231-8.12.3.1 Removal by Machining. Old blading should be machined out, if practical, to avoid distorting the dovetail. After the blade vanes and shroud sections have been removed by an axial cut across the top of the platform, a radial groove is machined into the roots along their centerlines to split them and relieve the interference fits. Take care to avoid damaging the rotor dovetail. Stop the cut just short of the interface, and use a hammer and chisel to split the remaining root material.

231-8.12.3.2 Removal by Driving. In cases where only part of a blade row is to be removed or if, for other reasons, machining is unfeasible, the blades may be removed by driving. First, the shroud bands shall be either removed or cut between each two blades. Remove the locking piece, loosen the blades one at a time, and drive them along the rim a few inches. Use penetrating oil to help free the blade roots. When two or more blades have been moved drive the remaining blades along the rim together, the number depending on the tightness of the fits. Using this method, there is less likelihood of damaging the rotor by canting the blades during the driving out operation. The dovetail should be continuously lubricated with a thin mixture of white lead and oil or castor oil (for saturated steam inlet turbines, white lead should be used by itself). Remove the blades by driving both ways from a position opposite the admission slot to reduce the length of arc through which each blade must pass. An air hammer may be used, but keep the tool clear of the wheel at all times.

231-8.12.4 PREPARATION OF ROTOR FOR NEW BLADES Methods used to prepare rotors for new blades depend on the rotor condition.

231-8.12.4.1 Rotor in Good Condition. If the rotor is not badly corroded and the old blades were carefully removed, the rotor should not require extensive preparation. These steps should suffice:

a. Remove burrs by careful filing.

b. Clean off corrosion using a wire brush.

c. Wipe the dovetail clean.

d. Apply a light coat of castor oil.

231-8.12.4.2 Rotor in Poor Condition. If the dovetail has become distorted or deteriorated to such an extent that buckets are loose when in their correct position, determine the location of the area of loose fit. When this is adjacent to the blade entry slot, the additional stock on the blade roots should permit fitting. In those cases where the distorted portion is located between two sections of the rim having a correctly sized dovetail, refer the problem to NAVSEA for a decision as to the advisability of cutting a new entry slot at the position of the loosest fit. Satisfactory results are seldom obtained by attempting to take a light cut off the high spots of the entire dovetail since the surface metal is usually quite hard. Removing material from the dovetail by hand methods is not recommended since this may upset the dimensional relations between the various load-carrying surfaces.

231-8.12.5 INSTALLING NEW BLADES. Considerations for installing new blades are described in paragraphs 231-8.12.5.1 through 231-8.12.5.5.

231-8.12.5.1 Blade Preparation and Fitting. Blades designed for installation with an interference fit on the rotor may have been stocked with excess material on the areas of fit. Carefully file and fit one blade to produce a light-drive fit on the rotor as specified on the applicable plan. Chamfer the edges of the root-fitted areas to prevent gouging the rotor. With lead-free lubrication, drive this blade completely around the rotor dovetail to determine whether the fit is satisfactory at all blade locations. Make a gage from this first blade so that succeeding blades can be fitted and checked with the gage before attempting installation. More than one gage may be necessary if the dovetail varies in size around the circumference. When machine tools are available, the blades can best be milled to fit the gages. In an emergency blades other than the first may be hand-fitted. When only a small number are to be replaced, making fixtures and assembling tools are impractical. Both the leading and trailing edge of those root surfaces that are tight against the rotor dovetail shall be chamfered with a file an amount not to exceed 0.030 inch. This will prevent gouging the rotor by the leading edge and will prevent a burr from protruding and interfering with the next blade in case the trailing edge metal pulls slightly when driven.

231-8.12.5.2 Weight Balancing Large Blades. Large blades (over about 10 inches in vane height) should be weighed and assembled in groups of equal weight. The heavier blades should be equally spaced from one another around the circumference of the rotor.

231-8.12.5.3 Blade Insertion. Force each blade into place by turning the rotor on centers or bearings, using a jig for supporting the blade at the correct angle throughout its length. This prevents objectionable vibration and minimizes the chance of damage to blades or rotor. Always lubricate rotor groove and blade root during installation. Castor oil mixed with graphite is recommended for this purpose. Do not use a lubricant containing lead. It may be necessary to replace blades by hand driving. An air hammer can be used to advantage, but in any case, the tip of the blade should be held in the hand by the person doing the driving, or by a helper. When driving use a brass drift (straight for inside root blades and pronged for the inverted type). The driving points must act against the root of the blade. Even though a brass drift is used, there may be some blade distortion. The root of an old blade loosely fitted between the drift and the blade being driven will minimize that distortion. In driving blades by hand, success has been obtained with special tools made to hold and protect the blades while being driven.

231-8.12.5.4 Blade Distribution. Drive blades both ways from the opening, thus reducing to a minimum the arc length through which each blade must pass. This requires that the first blade installed be positioned so that when all blades are installed those adjacent to the entrance opening will be properly positioned. Fill the wheel to within about 5 inches of the admission slot. Carefully select the remaining blades, interchanging blades of varying thicknesses so that the last blade installed on each side of the admission slot will overlap the edge by the amount shown on the wheel assembly plan, usually about 1/32 inch. When necessary to remove root material to accomplish this fitting, take a small amount from five or six blades, rather than a large amount from one or two.

231-8.12.5.5 Locking Piece. Insert tapered drift(s) between the two blades adjacent to the entrance opening, and drive so as to force the blades apart. Fit, assemble, and secure the closing or locking piece in accordance with the wheel assembly plan. On wheels with outside dovetails, the closing piece, or closing blades, will be pinned and the new piece will cover the existing hole or holes in the wheel rim. To facilitate locating such a hole, establish two prick punch references on the wheel so that the distances between the two punch marks and between each punch mark and the center of the hole are equal. With the new closing piece in place, set dividers to the distance between punch marks and scribe two arcs on the closing piece, using each punch mark as a center. The intersection of the two arcs is the hole location. It may be possible to use original size locking pins; otherwise, reaming out the hole may require machining oversize pins to fit size on size.

231-8.12.6 INSTALLING SHROUDING. Blades are normally designed with round cross-sectional tenons, except where blades are too thin to permit a round tenon with the necessary area. Cut shroud band material from stock to the proper lengths, according to the specified blade grouping, and add tenon holes.

231-8.12.7 TEMPLATE. Lay a template of card stock or similar material, the length of a shroud band, along the blades adjacent to the blade tenons. The spacing of the tenons should be accurately marked on the template. Bend each shroud band to the curvature of the blading. Lay the template on this curved shroud, and accurately transfer the markings from the template to the shroud. Do not transfer the markings made on a curved template to the shroud when it is flat because the holes will not line up with the tenons.

231-8.12.8 TENON HOLES. Tenon holes may be drilled, milled, or punched. Drilled or milled shrouding does not require annealing. Shrouding of 12-chrome steel to be hot punched requires annealing at 1,500°F to 1,650°F and slow cooling before punching. After punching, heat treat at 1,800°F to 1,850°F, quench in oil, and temper at 1,100°F to 1,150°F to obtain a hardness of 160 to 240 Bhn. Shrouding of 12-chrome steel to be cold punched in the heat-treated condition (above 160 Bhn hardness) must be stress relieved after punching by heating to 1,100°F, holding for 2 hours, and furnace cooling to 800°F. Assemble the shroud bands on the blading, and cut the lengths in accordance with the wheel assembly drawing. Neither deflect blades nor elongate holes to make the band fit. If the holes and tenons do not line up, prepare a new band.

231-8.12.9 PEENING TENONS. Some vendors are using automatic machines to peen tenons. The tenon is heated to a specific value for the type of material, and the head is formed in a way that eliminates stress risers. Considerations for peening tenons are discussed in paragraphs 231-8.12.9.1 through 231-8.12.9.4.

231-8.12.9.1 Peening Tools. Riveting tools should be used for riveting tenons, but this work may also be done readily with a peening hammer. The peening should be done around the edge of the tenon, rather than directly on the top. The tenon should be swelled enough at the top to fill the clearance between the hole in the shroud and the tenon. The riveted tenon is subjected to tensile stress and cyclic bending stresses if vibration is present. It must be free from stress raisers (such as cracks or folds) that could result from excessive peening. Peen each banded group by starting at the center blade and working alternately to each end. Some tenons are designed to be heated for peening (refer to the applicable drawings). Do not repeen tenons.

231-8.12.9.2 Blocks and Wedges. Take particular care when peening tenons of blades having relieved backs. Fit hardwood blocks to the concave side of each blade in one section, and insert wedges between these pieces and the backs of adjacent blades. Insert the blocks and wedges at the point of smallest cross section so that maximum support and rigidity can be obtained over that section. These blocks can be used progressively around the wheel as the peening is done and are necessary to prevent bending, as would be the tendency with this type of blade if it were unsupported.

231-8.12.9.3 Angled Shrouding. Take similar precautions when peening the tenons on blades where the blade tip is slanted and makes an angle with the axis of the blade. In this case, the shroud is shaped to conform to the angle of the blade tip before it is assembled. During peening, this angle imposes a side strain on the blade root. Avoid this strain by blocking to an adjacent wheel.

231-8.12.9.4 Shroud Machining. After installation, the shrouding will be machined as necessary. Dimensions can be obtained from the applicable assembly drawings. Since machining variations are sometimes encountered, obtain the clearances before removing the rotor from the turbine. Also, obtain the dimensions of the original shrouding relative to an appropriate reference before removing the blading. Considering this data will permit machining the new shrouding to obtain optimum clearances.

231-8.12.10 REPLACING DAMAGED AXIAL ENTRY GROUPS. Damaged axial, or side-entry, blades may be repaired by removing and replacing only those blade groups that contain damaged blades. To accomplish this, first consult the applicable technical manual and the wheel assembly drawing for special instructions and determine the method of locking the blades. In general, the shroud band on the group to be removed must first be either removed or sawed through between each blade. Next, saw off the trailing edge of the trailing blade from the group since it overlaps the leading blade of the adjacent group. The saw cut should extend down through the blade platform if the platform forms a lock by overlapping the adjacent platform. Depending on the design of the platform or locking key, removal of material by drilling may be required to free the first blade. Take care to avoid damaging the adjacent blade or the wheel. When the first blade is free, that blade and the rest of the group can be driven out. Install the new blades, and modify the last one by removing the interfering part of the vane trailing edge and the locking portion of the platform. The modified blade must be ground smooth and have the trailing-edge radius restored.

231-8.13 DYNAMIC BALANCE OF ROTORS

231-8.13.1 GENERAL. To ensure a smoothly operating unit, the propulsion turbine and SSTG turbine manufacturer performs many balancing operations on the rotor in the manufacturing process. These operations permit correction for imperfect geometry and dissimilarity of materials at the particular plane in which they occur. The completely assembled rotor is usually hotbox checked for balance at elevated temperature and overspeed (parts may shift under these conditions) and may be additionally in-place balanced at load if required by specification.

231-8.13.2 HIGH-SPEED BALANCING. High-speed balancing is not really a fine tuning of low-speed balancing because of different mode shapes. Most manufacturers final balance their rotors at and above operational speed in a high-speed balance facility that operates in a vacuum. A high-speed balanced rotor may well be out of balance at low speed. No rebalance shall be made because of the difference in mode shapes. Turbine rotor balance should be checked whenever work is done on any rotating element. Since most repair activities have only slow-speed balancing facilities, rotors that have not been reworked or do not have reported vibration problems should not be balanced without consulting the OEM. Knife-edge rollers shall not be used against chrome-plated journals when high-speed or low-speed balancing a rotor.

231-8.13.3 ALINING TURBINE WITH DRIVEN GENERATOR. The most important alinement in a horizontal turbine relates to the coupling between the turbine and the driven component. The shafts of the two must be alined correctly. It is very important, therefore, to determine that the face and sides of the coupling run parallel and true to each other. Alinement may be checked with the coupling installed or removed, whichever method has been used successfully by the repair activity.

231-8.13.3.1 The preliminary alinement is usually made when the apparatus is cold and before piping has been connected. Bear in mind that the turbine centerline may or may not rise when steam is turned on, depending on the direction of rotation; that is, whether the pinion teeth push down or lift up in rotating the bull gear. For this reason always consult the manufacturer's plans and instruction manual for alinement data. The final alinement should be made after the turbine has been connected to the piping and the ship is waterborne. The final alinement should be such that the coupling will not run out; otherwise, vibration will result and the bearings will wear rapidly.

231-8.13.4 DISCUSSION OF STATIC AND DYNAMIC BALANCE. Imbalance in any turbine rotor can be considered to consist of static and dynamic components. Static imbalance is the eccentric weight that will cause a rotor to turn, when supported at its bearings on horizontal straight edges, so that the heavy side will move to the bottom. The dynamic component is that portion of the imbalance that will not produce any motion on straight edge supports but will tend to induce circular motion of the rotor ends when in operation. End motions, at any instant in time, would be in opposite directions, while the rotor midpoint would experience no displacement. In force terms, pure dynamic imbalance would consist of equal and opposite radial forces (centrifugal effect of eccentric masses) acting at points axially displaced along the rotor longitudinal centerline equidistant from the rotor's midpoint in the central transverse plane.

231-8.13.5 REASONS FOR DYNAMIC BALANCE. To obtain perfect balance, the inertial or mass centerline of the rotor would have to coincide exactly with the rotational centerline. Since this ideal is unattainable, practical approaches to balancing are to limit centrifugal forces (function of residual mass imbalance, radial distance at which it acts, and speeds) to those that will not damage supporting bearings or produce alternating stresses that will cause fatigue failure of rotor parts. In special cases, balance of rotors is further refined by in-place balancing to minimize structureborne vibrations.

231-8.13.6 PRINCIPLES INVOLVED IN BALANCING. Basically, balancing consists of adding weights, in previously established balancing planes of the rotor, that will exactly counterbalance existing rotor imbalances. Balancing should be performed only by qualified personnel with an established reputation for successful balancing.

231-8.13.6.1 The force resulting from imbalance has magnitude and phase, which must be sensed and read out on suitable equipment. Whether balancing is to be accomplished on a balancing machine or in-place with portable balancing equipment, the basic steps required are:

1. Obtain readout of force, or vibration proportional to force, and phase in the rotor as-is condition at highest practical or permissible speed.

2. Rerun at the same speed with a trial weight in a balancing plane to determine the effect of weight on imbalance.

3. Similarly, obtain additional influence information from trial weights installed, one at a time, in other balance planes.

4. Either compute by simultaneous equations the weight and angular placement of weight in each plane for balance or consider, on a single-plane basis, the effect of weights in each plane on all force or vibration sensors and apply appropriate weights successively in one plane at a time to optimize balance.

5. Permanently install balance weights and verify that balance is within specification at balancing speed and over the normal speed range of the unit. Do not drill or grind on rotor to remove weight for balancing without specific approval by the turbine manufacturer and NAVSEA. The relative merits of the various balancing methods presently available are compared in Table 231-8-4.

231-8.13.7 ACCEPTABLE LIMITS OF IMBALANCE. Limits of imbalance are normally specified in two ways:

a. In terms of the permitted amount of residual imbalance in appropriate units (such as ounce-inches) that can be determined only in the actual balancing process

b. The effects of imbalance on rotor or casing vibration that can be checked with portable vibration equipment.

Table 231-8-4 RELATIVE MERITS OF BALANCING TECHNIQUES

Low-Speed Balance Machines, Portable Type		Low-Speed Balance Machines, Shop Type		High-Speed Balance Machines, Shop Type		In-Place Balancing	
Advantages	Disadvantages	Advantages	Disadvantages	Advantages	Disadvantages	Advantages	Disadvantages
1. Can be used aboard ship. Large rotors need not be removed from ship.	1. Usually only very low (approximately 100 RPM) speed permitted due to rigging of drive and large rotors involved.	1. In shop, rotor can be balanced in shortest time because time and movement are minimal.	1. Machine any be out of calibration.	1. 2. Short balancing time.	1. Few machines available at major turbine manufactures.	1. All effects of load speed, and thermal effects are present.	1. Time consuming, may take up to 4 hours or more because of requirement for stabilization, securing, and startup following weight changes.

Table 231-8-4 RELATIVE MERITS OF BALANCING TECHNIQUES - Continued

	2. Too sensitive to meet normal balance requirements for propulsion turbine rotors.	2. Generally will produce mechanically suitable balance.	2. A rigid rotor balance may be unsuitable for flexible high<#0106>speed rotors.	2. Greater choice of bal<#0106>ancing planes that can be used effectively to compensate for flexible rotor effects.	2. Machine may be out of calibration.	2. Produces lowest vibration levels.	2. Requires that ship be operated exclusively for balance work.
			3. Will not correct for heat and load effects.	3. Speed of balance closer to the machine operating speed.	3. Same as 3 and 4 for low<#0106>speed balance machines.	3. Eliminates errors that can be introduced by balance machine drives.	3. To achieve the low vibration potentials, balanc<#0106>ing must be done by experienced personnel with appropriate equipment.

Table 231-8-4 RELATIVE MERITS OF BALANCING TECHNIQUES - Continued

			4. Coupling turbine to driven unit may cause imbalance.			4. Merits of balancing effort can be checked independently of balance equipment with instruments.	4. Limited planes available for weight addition and amount of weight that can be used.

231-8.13.7.1 The allowable residual imbalance for almost all propulsion turbine rotors is shown in MIL-STD-167 (SHIPS), Mechanical Vibration of Shipboard Equipment, as

$$U = \frac{4W}{N}$$

Where

U = maximum allowable residual imbalance in each balance plane in ounce-inches

W = weight of rotating part in pounds

N = maximum operating RPM of unit.

231-8.13.7.2 The allowable turbine bearing cap vibrations are generally in accordance with Figure 231-8-13. In this figure, ″PAT″ refers to preliminary acceptance trials for a ship and ″FAT″ refers to final acceptance trials for a ship. Figure 231-8-14 is a vibration nomograph to be used when converting AdB to displacement in mils peak to peak. High vibration levels are often caused by other than rotor imbalance. Accordingly, before making a decision to balance a turbine rotor, perform the following inspections or checks (follow with a detailed vibration survey or analysis):

a. Confirm that all turbine and pinion bearings are in satisfactory condition and that bearings are correctly alined with the shaft.

b. Confirm correct alinement between turbine rotor and pinion.

c. Confirm that coupling is in satisfactory condition.

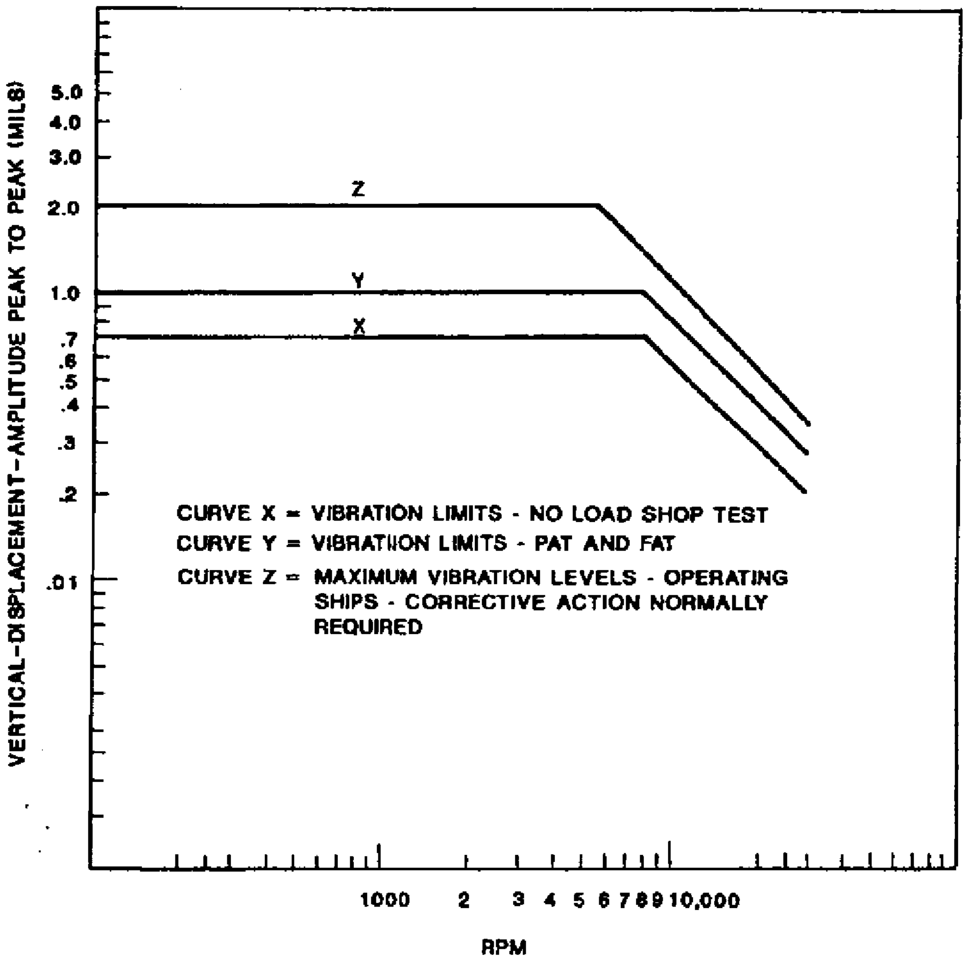


Figure 231-8-13 Steam Turbine Vibration Limits - Bearing Cap

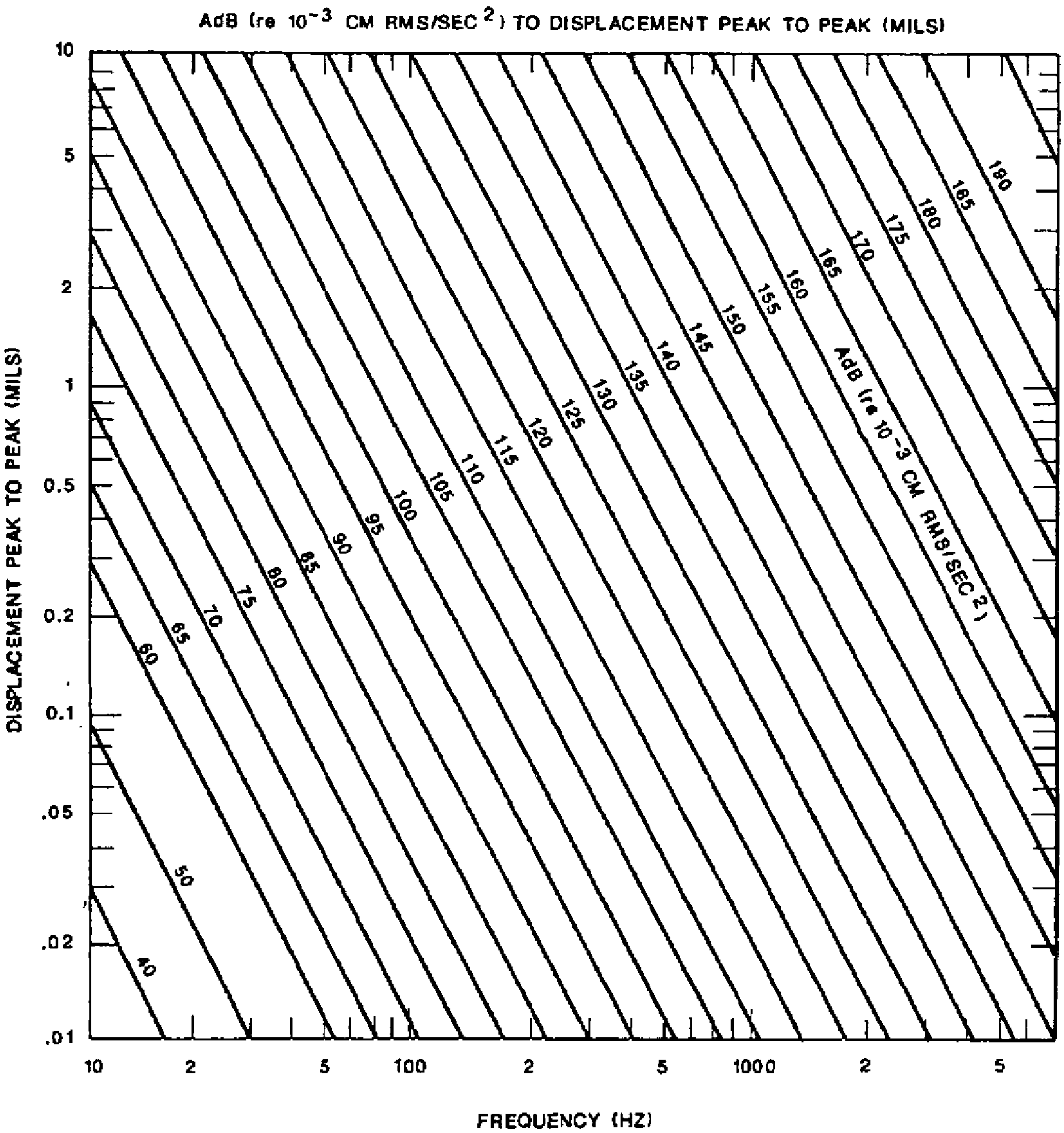


Figure 231-8-14 Vibration Nomograph

231-8.13.7.3 Information in the applicable turbine technical manual concerning vibration and balancing should also be used as a guide. Inexperienced personnel should not attempt balancing.

231-8.14 REBABBITTING OF BEARINGS

231-8.14.1 GENERAL. Manufacture of bearing shells or backings is the largest part of bearing cost. This is especially true in the case of spherically seated bearings because of the complex machining operations required and the close tolerances. Bearings that are no longer serviceable because of surface damage or excessive clearance should therefore be rebabbitted by a qualified repair facility whenever possible in preference to procuring new bearings. Tilt pad journal bearing pads and thrust bearing pads are not economical to rebabbitt. Procure replacement pads as needed and scrap the old ones.

231-8.14.2 SPARE BEARINGS. Ships normally carry enough spare bearings to replace each bearing in one turbine or one SSTG. Supply centers and tenders also stock replacement bearings. Use these to replace failed bearings. Return the failed bearings to the supply system to be rebabbitted by a qualified depot level repair facility. Qualification is described in DOD-STD-2183, Bond Testing, Babbitted-Lined Bearings, and DOD-STD-2188, Babbitting of Bearing Shells (Metric).

231-8.14.3 LIMITING OUTSIDE DIAMETER AND INSIDE DIAMETER RATIO FOR REBABBITTING. Before a bearing shell is rebabbitted, it should be inspected for condition and suitability. Relatively thin shell bearings (that is, those having a ratio of shell outside diameter to bearing inside diameter (OD/ID) of 1.25 are not recommended for rebabbitting. This type bearing should normally be replaced because, in most cases, rebabbitting attempts will result in faulty bond or excessive distortion.

231-8.14.4 GEOMETRIC CHECKS BEFORE REBABBITTING. A bearing shell considered suitable for rebabbitting should be inspected for dimensional conformance with the manufacturers' drawings and for the condition of the seating surface. In some cases nicked and dented seating surfaces can be repaired for example, bearings shells with removable pads that form the spherical diameter. A similar method of repairing shells without pads is to spread the shell slightly at the joint split and build up the joint faces with weld to increase the outside diameter of the shell and permit remachining of the seating surface. The latter is considered expensive and therefore restricted to emergency situations.

231-8.14.5 PROCEDURE. Bearing shells should be rebabbitted in accordance with DOD-STD-2188, which covers preparation of the bearing shell and casting the babbitt. Bond testing shall be performed in accordance with DOD-STD-2183. Although both static and centrifugal casting are generally acceptable for producing a satisfactory metallurgical bond, centrifugal babbitt casting is preferred. No pits visible to the naked eye and normal lighting are allowed in the finished internal diameter of the journal bearing or in the finished surface of the thrust bearing shoe. Such pits would interfere with future troubleshooting, such as inspecting for electrostatic discharge or cavitation damage. Proper static or centrifugal casting should prevent pitting. The babbitt metal used is FED Spec QQ-T-390, Grade 2 (preferred) or Grade 3.

231-8.14.5.1 Pay particular attention to special features when machining the bearing bore. Use the manufacturer's drawing as guidance, and locate and accurately machine all details such as elliptical bore, babbitt relief, and grooving. The bore diameter can be altered to provide the proper clearance when the bearing is to be used with an undersize journal.

231-8.14.5.2 All resistance temperature elements (RTE's) shall be installed by qualified (see paragraph 231-6.1.3.3) RTE installers or under the direct supervision of a qualified technical representative currently employed by the original turbine manufacturer. If RTE's must be installed in a bearing, use the following procedure:

1. First test it per paragraph 231-6.4.7.3.
2. Install wires through the hole in the babbitt, pulling out the bearing back gently until the RTE is seated against the shoulder in the hole.
3. Maintain pressure gently on the wires while puddling in the correct babbitt over the top of the RTE.
4. Form a raised button of babbitt on the bearing surface. When it is cooled, scrape in the button flush with the babbitt surface.

5. Use the correct epoxy potting compound to secure the wires in the groove in the back of the bearing. So that the wires won't be cut, round off any sharp corners in the groove in the back of the bearing or the groove in the bearing bracket.
6. Apply putty in several spots to hold the wires in the groove. Putty may be boiler sealing compound MIL-S-17377, Sealing Compound Boiler-Casing, NSN-8030-00-264-388 8 or equal.
7. Apply the correct epoxy potting compound to secure the wires in the bearing groove. Let it cure.
8. Remove the putty and fill in the voids with more epoxy.
9. Contour the epoxy flush with the bearing back with a file as necessary.
10. Route the wires leaving the bearing in the bracket to prevent damage. The wires may be contained in the bearing bracket groove with an appropriate silicon compound such as RTV-108.
11. When wires are installed near couplings or other rotating components, keep the wire lengths short and captive away from rotating fasteners or teeth. Loop excess wire lengths and secure well with tie-wraps or appropriate fasteners.
12. Solder the wire ends to the terminal block as necessary.

231-8.14.6 BABBITT. In the past, bearings were bought with babbitt material that met FED Spec QQ-T-390 Grade 2 or 3 requirements. Grade 2 babbitt material is preferred. It has a hardness of 24.5 Brinell at 68°F and 12.0 Brinell at 212°F; is solid below 466°F and liquid above 669°F; and is putty-like between 466° and 669°F. Grade 2 is 88 to 90 percent tin, 7 to 8 percent antimony, and 3 to 4 percent copper. For further details, see FED Spec QQ-T-390.

231-8.15 RECONDITIONING BEARING JOURNALS

231-8.15.1 DRESSING UP MINOR JOURNAL BLEMISHES OR SCRATCHES. Time and money are too often wasted reworking rotor journals for appearance purposes with, at best, questionable improvements in performance. In many cases, such as submarine-turbine rotors, where journal roundness is quite critical, reworking for appearance reasons is prohibited because performance and noise levels can be seriously compromised. Only experts qualified in reconditioning turbine journals should dress or repair imperfections. A qualified (see paragraph 231-6.1.3.3) expert may strap-lap a journal with a 360-degree wrap of crocus cloth. This has successfully removed the frosting caused by electrostatic pitting and has removed the high metal from journal phonographing (see Table 231-6-3) without reducing the journal diameter below specification. Do not hand-stone journals. See Table 231-6-4 for guidance on journal inspection.

231-8.15.1.1 The following conditions, to varying degrees, will be found on most turbine journals:

a. Scratches, both circumferential and axial (scores, lines)
b. Pits (depressions, dents)
c. Discoloration (blemishes, shiny areas, dull areas).

231-8.15.1.2 In most cases, these conditions do not justify any corrective action. When the number and size of scores or pits are excessive, the bearing will experience rapid weardown of babbitt. That is, if the bearing associated with the questionable journal has no replacement history due to wear, the condition can be judged satisfactory and no repair action should be attempted.

231-8.15.1.3 If journal surface conditions are causing rapid weardown or pickup of babbitt, a turbine manufacturer's representative or a NAVSEA-designated representative should inspect the journal and determine the correct in-place repair procedure to be used.

231-8.15.1.4 Journal reconditioning or repair can and should be done in-place in the ship, except in cases where the rotor has been removed from the ship for other repairs. Approved repair procedures in the order of preference are:

a. Hone, or grind and hone in-place to correct surface conditions. Install standard bearing if diameter is reduced by 0.003 inch or less below designed diameter. Install special bearings if reconditioning of journal results in reducing the diameter by more than 0.003 inch below designed diameter.

b. The same procedure as discussed in item a above shall be done in a shop if the rotor has been removed from the ship for other reasons.

c. Machine journals to correct unsatisfactory conditions, and chrome plate to design diameter. Plating shall be done off the ship (see paragraph 231-8.15.6).

231-8.15.1.5 Do not repair rotor journal surfaces by metal or plasma spray, or by flame spray.

231-8.15.2 PRECAUTIONS WHEN DRESSING A JOURNAL IN-PLACE. Take the following precautions to protect the bearing babbitt and oil passages from foreign material:

a. Remove lower half of bearing. (When working on journal, support the shaft in special dummy bearings.)

b. Plug oil supply and drain passages. Rags, paper, or plastic plugs may be used for this purpose. Do this in such a way that the plug may be easily removed when it has served its purposes. Removal is mandatory.

c. After completing repair, wash the journal and interior of bearing housing with cleaning solvent. Wipe dry with lint-free rags to remove all traces of abrasive material.

d. Remove all oil passage plugs before bearing reassembly.

e. Do not use Teflon blocks or wedges to support rotor journal when repairing.

231-8.15.3 JOURNAL RECONDITIONING THAT REQUIRES UNDERSIZE BEARING. When the journal repair requires special bearings, the repair activity shall:

1. In addition to installing a nonstandard bearing, provide or rebabbit and bore one ship spare bearing to suit the undersized journal. Properly tag the spare bearing for identification, and carry as an onboard repair part (in excess of allowance if necessary).

2. Provide and install a suitable label plate on or immediately adjacent to the bearing cap stating that the journal has been machined undersize and a nonstandard bearing is required in the event of replacement.

3. Notify NAVICP, Mechanicsburg, PA, of special bearing requirement and provide a unique OEM part number.

231-8.15.4 LIMITATIONS ON AMOUNT OF JOURNAL METAL REMOVED. In general, reduce the journal diameter following the guidelines in paragraphs 231-8.15 through 231-8.15.1.5. If several journals that could normally use the same spare bearing are to be machined, however, reduce the journals to a common diameter to minimize the number of nonstandard bearings required for onboard spares. Without NAVSEA approval, no journal shall be reduced in diameter by more than 1/4 inch or beyond a diameter that will increase the torsional shear stress 25 percent above the original design, whichever occurs first. Established limits are provided in Table 231-8-5 and the formula for determining the minimum journal requirements is shown in Figure 231-8-15.

Table 231-8-5 LIMITING JOURNAL REDUCTION

Type Shaft	Original Design Diameter	Minimum Diameter to Which Shaft May Be Reduced
Solid	Less than 3.6 inch	93% or original design diameter
Solid	3.6 inch or greater	Original design diameter less 1/4 inch
Bored	Any	In accordance with Figure 231-8-15 (formula and note)

231-8.15.5 PROCURING NONSTANDARD BEARINGS. Nonstandard bearings are sometimes called undersized or oversized bearings, but the term nonstandard bearing is preferred. Nonstandard propulsion turbine bearings may be available from stock by NAVICP, as ordered by the fleet or repair activities. Undersized bearings are assigned a separate National Stock Number (NSN) completely different from that assigned to the comparable standard bearing. Once established, these NSN's are added to the applicable allowance lists.

231-8.15.5.1 Undersized bearings, for which the stock number is unknown or unassigned, may be ordered by listing the following information:

a. Identity of ship for which the bearing is being procured.

b. Identity of the bearing by component identification description (CID) number of the component in which it is to be used, and the NSN and nomenclature of the standard bearing taken from the allowance list (plus any additional wording required to properly identify the bearing).

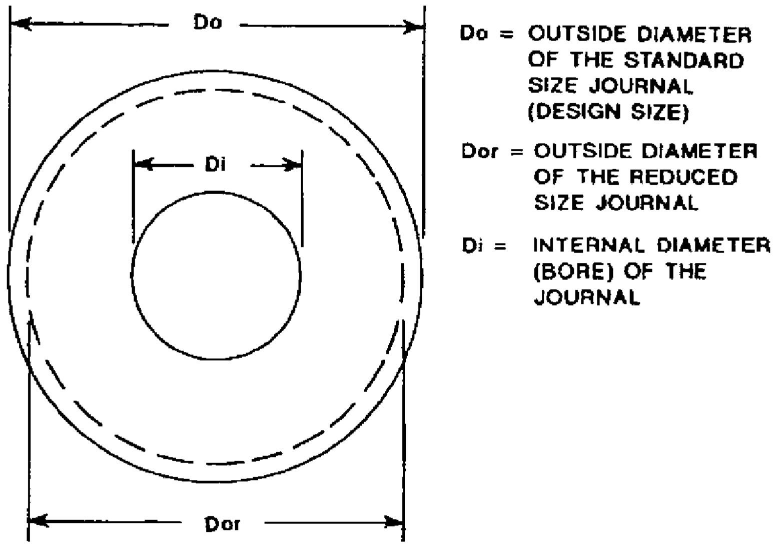
231-8.15.5.2 An example of proper identification is:

CID XXXXXXX NSN XXXX-XXX-XXXX BEARING-LP-END STBD HP TURBINE

231-8.15.6 DEPOSITING NEW MATERIAL ON JOURNAL. Worn bearing journals may be restored to original design size by chromium plating. Because of the extensive service and reliability required of steam turbine bearing journals, guidelines are set forth as to the type and permissible buildup by plating. Plating for bearing journals shall be of the industrial hard-chrome applied by the dip or tank method. Brush plating has not been approved for bearing journals. Metalizing bearing journals by spray welding is prohibited.

231-8.15.6.1 Chrome Plating. Requirements and recommendations for chrome plating are given in QQ-C-320 CL2, Chromium Plating (Electrodeposited), or DOD-STD-2182, Engineering Chromium Plating (Electrodeposited) for Repair of Shafting (Metric). Unless certain precautions are exercised, chromium plating is conducive to premature failure of rotating steel shafts. Surface prestressing in compression (such as produced by shot peening, cold rolling, or nitriding) is effective in sustaining resistance to fatigue. Some of the basic requirements considered essential for turbine bearing journal plating are described in paragraphs 231-8.15.6.5 through 231-8.15.6.7.

231-8.15.6.2 Preliminary Inspection. Check the rotor for straightness and concentricity by indexing. The rotor should be within specified tolerances.



$$\left(\frac{Dor}{Do}\right)\left(\frac{Do^4 - Di^4}{Dor^4 - Di^4}\right) \leq 1.25^*$$

NOTE: IF THE SELECTED VALUE OF Dor SATISFIES THIS EQUATION, THE SELECTED VALUE OF Dor IS SATISFACTORY, PROVIDED IT IS NOT MORE THAN 1/4 INCH LESS THAN Do.

*THE SYMBOL ≦ MEANS LESS THAN OR EQUAL TO.

Figure 231-8-15 Formula for Determining Minimum Journal Requirements

231-8.15.6.3 Blemish Removal and Reducing Shaft Diameter. Completely remove all grooves, pits, and other blemishes from the area to be plated. Final metal removal for diameter reduction should be by wet grinding. The reduced diameter should be uniform throughout the entire length, except at fillets, and concentric with the shaft axis. The length of the reduced diameter section should extend 1/8 inch beyond both ends of the bearing surface.

231-8.15.6.4 Shot Peening. Unless otherwise approved by NAVSEA, journal surfaces to be chromium plated shall be shot peened with Number 170 or smaller cast steel shot to an intensity of 0.012 to 0.016 inch, as indicated by an A-gage strip (specified in MIL-S-13165, Shot Peening of Metal Parts), as a surface prestress treatment before plating.

231-8.15.6.5 Plating Quality. Chromium plating shall be of the type variously designated as Bright, Hard, Engineering, or Industrial. Regulate and control the plating process to produce deposits that are fine grained, adherent, free from milky (or frosty) areas, visible blisters, pits, nodules, porosity, and burning as a result of excessive grinding wheel pressure or current density. After finish grinding, the plating should blend smoothly with the adjacent unplated surface without depression at the plating boundary. Visible crazing is acceptable after finished grinding. Therefore, liquid penetrant testing should not be done on this surface as an acceptance test. Deposit the plating directly on the steel surface without a metallic undercoat. If the plated surface becomes damaged in service, visually inspect the damage for acceptance and test adherence by removing oil residue with an appropriate solvent; then quickly pull a length of very sticky tape from the surface. Any flakes of chrome are cause for rejection.

231-8.15.6.6 Plating Thickness. After finished grinding, the thickness of chromium plating shall be the minimum necessary to meet dimensional requirements and correct the deficiency, and within the following limits:

a. After finish grinding, 0.0015-inch minimum radial thickness

b. After finished grinding, 0.025-inch minimum radial thickness.

231-8.15.6.6.1 Where these values are to be exceeded, refer the matter to NAVSEA for approval before processing the shaft.

231-8.15.6.7 Maximum Increment of Deposition. To avoid deleterious effect of modulation and to permit maximum rate of deposition (0.012-inch radial thickness), finished radial thicknesses in excess of 0.010 inch shall be applied in approximately equal increments and each increment ground to remove from 0.0005-to 0.002-inch radial thickness of chromium.

231-8.16 ROTOR PACKING LAND RECONDITIONING

231-8.16.1 REPAIR. Experience has shown that repair to the packing land areas of turbine rotors may be required after 10 to 15 year of operation. When the repair is required, the following repair approach is recommended:

1. Machine rotor lands (both high and low) to achieve a satisfactory condition.
2. If machining results in a reduction in diameter from design of 0.005 inch or less, install standard packing rings.
3. If machining results in a reduction a diameter of greater than 0.005 inch, install special underside packing rings.

231-8.16.2 NAVSEA APPROVAL. For a reduction in diameter from design of greater than 0.100 inch, repair procedure and approval shall be obtained from NAVSEA on a case basis. Metal spray, flame, or plasma spray are prohibited. Rotor welding is allowed only if a NAVSEA-approved welding procedure is used. Chrome plating to MIL-STD-2197 is permitted on flat packing lands. Chrome plating is prohibited on stepped or high-low packing lands.

231-8.17 RECONDITIONING THRUST COLLAR

231-8.17.1 MACHINING. If a thrust collar becomes badly scored, the scored surface can be machined and refinished. As a rule of thumb, the thrust collar design thickness can be reduced 10 percent by machining. See paragraph 231-6.1.3 for technical assistance. The collar can be set up in a lathe and machined, using an ordinary cutting tool or tool post grinder. Take care during setup and machining to ensure that the thrust faces remain square with the bore in the collar. Check the remachined collar for surface flatness and uniform thickness by measuring the collar thickness at various locations. Thickness should not vary more than 0.0005 inch. A surface finish as recommended by the manufacturer, typically 16 Roughness Height Rating (RHR), is required for thrust collar bearing surfaces, which is obtained by lapping or honing the machined surface to remove all tool or grinding marks. A completely smooth mirrorlike finish is undesirable since it will not allow the necessary oil film to form and the shoes will be destroyed.

231-8.17.2 CARE OF TAPERED-LAND THRUST BEARING SURFACES. If the babbitted thrust faces of the thrust plates or shoes of a bearing become bruised, remove the bruises by scraping the faces to an accurate surface plate. In the case of tapered land thrust bearings manufactured by General Electric, however, the tapered land thrust plate must be scraped using a special fixture provided. For a description and directions for use of the special fixture, consult the manufacturer's instruction manual. Slight rusting of the collar faces can be removed with a fine oil stone. If deep rusting occurs, refinishing will be necessary. Never use a coarse-grained stone, a scraper, or a file on the collar faces.

231-8.18 REPAIR OF CASING EROSION AND CORROSION

231-8.18.1 ACCEPTABLE REPAIRS. Considerable metal could be lost from susceptible, internal turbine surfaces when operated extensively with wet stream (paragraphs 231-7.5 through 231-7.5.4.3). Restoring of casing surfaces by welding and remachining at the manufacturer's plant, or at a qualified repair activity, is an acceptable repair technique. The qualification aspect of a repair facility is stressed since welding must be controlled to keep distortions within those correctable by machining. Also, the precision of the machine work on seal and clearance establishing surfaces must be equivalent to that of the original manufacture.

231-8.18.1.1 Minor Weld Repairs. Repair activities shall, on a case basis, obtain NAVSEA approval to perform welding on alloy steel steam-turbine casings, or major weld repairs to casing cracks, porous areas, or diaphragms. Minor steam cuts in carbon steel horizontal joints or steam chest joints may be weld-repaired without NAVSEA approval.

231-8.18.1.2 Approval Needed. In the past, yards have proceeded with turbine welding repairs without approval, only to have to remove the entire casing from the ship for stress relieving later. By obtaining advance approval from NAVSEA, this could have been avoided.

231-8.18.1.3 Arc Strikes. Arc strikes are hardened metal flaws that can occur when a weld rod touches metal or when certain designs of magnetic particle testing yokes are used. Arc strikes on 12-percent chrome turbine surfaces should be:

a. Removed by grinding

b. Etched to find residual flaw

c. Surveyed for microhardness of affected area

231-8.18.2 CLAD-WELDING OF CASING. The use of corrosion resistant (stainless steel) materials as overlays on carbon steel greatly improves the resistance to wet stream erosion. The typical repair calling for this clad-welding type would require that the turbine be removed from the ship. Policy requiring NAVSEA approval for all welding on turbines is discussed in paragraphs 231-8.1 through 231-8.3.

231-8.18.2.1 Procedure. The major steps in the procedure would be:

1. Inspecting to ascertain the extent of damage and which parts require repair

2. Establishing dimensional details of the fix

3. Machining parts preparatory to welding

4. Welding and rough machining
5. Finish machining
6. Final inspection

231-8.18.2.2 Inspection. In the inspection pay particular attention to the casing sealing surfaces where they contact the diaphragms, gland rings, and nozzle plate. Measurements in the inspection phase are necessary to determine existing dimensions that must be re-established to retain clearances with rotor and alinement with gear or condenser, if applicable. These would include:

a. Height of rotor centerline with respect to mounting feet plane(s)
b. Axial distances from any vertical joint to blading or diaphragm grooves
c. Hog or sag of base and cover

231-8.18.2.3 Repair. About 1/16 inch shall be machined off eroded surfaces. This will normally clean up irregularities and give adequate and uniform thickness of cladding material. Areas adjacent to welds are upset to avoid low spots from weld undercutting. Welding is performed with a MIL-E-16715, Electrolytes Welding, (Corrosion-Resisting Steel), type MIL-309-15 or MIL-310-15 class I electrode, in accordance with MIL-E-22200/2D, Electrodes, Welding, Covered Austenitic Chromium-Nickel Steel, using a skip-welding technique and peening of each pass to minimize distortion. Rough machining of all welding surfaces is limited to that which will leave 1/32 inch for finish machining. Final machining shall be such as to make all steam joints tight. Clearances and alinement surfaces shall be in accordance with initial inspection reference dimensions. The application of this particular cladding material requires no pre- or post-heat.

231-8.18.3 WELD REPAIR OR DIAPHRAGM SEAL SURFACES. Weld repair of nozzle diaphragms generally follows the casing-weld procedure. The diaphragms are somewhat more susceptible to welding distortions, and some experimentation may be necessary to determine the amount of distortion that will occur. This establishes the cold prework (upsetting, rolling, jacking) necessary to allow finishing of diaphragms to the desired dimensions. Strap-weld the two halves of the diaphragm together, and strap-weld the assembly to a machined plate. The vertical seal surfaces are rolled (upset) and welded first to permit truing of weld surfaces to the original diaphragm's inner ring bore. Welding and machining of the diaphragm horizontal joint is then accomplished referenced to the outer ring OD. Finish the diaphragms by milling necessary keyway slots in horizontal joints and by fitting keys and crush pins.

231-8.18.4 WELDING PROBLEMS. All welding on or around turbines requires properly grounded welding setups. Before striking an arc on a turbine or pipe or turbine component, verify proper grounding. Contact a welding engineer as necessary. If welding is done on a turbine without proper grounding, inspect all bearings and repair or replace if arcing occurred across the bearings. Also, before welding verify that you have NAVSEA approval as necessary (see paragraph 231-8.18.1.1). Verify the proper base material, proper welding procedure, and proper certification of welders. Unintentional arc strikes on an alloy turbine casing must be ground off and etched to prove removal. See paragraph 231-8.18.3.

231-8.19 BLAST CLEANING STEAM TURBINE INTERNAL PARTS

231-8.19.1 To prevent excessive surface roughening and base material removal, control of the cleansing agent and agent article size is essential when blast cleaning steam turbine internals (rotors, blading, casing internal surfaces). When choosing the type of blast cleaning material, consider the turbine manufacturer's recommendations. Be careful not to burn through surfaces of blades through intense, prolonged blasting in one area. On the basis of previous experience and equipment manufacturers' recommendations, the following guidance is provided:

a. One cleaning agent is aluminum oxide with a particle size no coarser than 220. Other cleaning agents, such as glass beads, are considered acceptable provided the finishing result is equivalent to that provided by 220 grit aluminum oxide. Do not use sand, ash, or walnut shells.

b. Before cleaning, tape over all rotor journals, thrust bearing surfaces, and metal-to-metal steam joints.

c. Rebalance rotor after blast cleaning, if required, in accordance with paragraph 231-8.13.

231-8.20 GOVERNOR PROBLEMS

231-8.20.1 TYPES OF PROBLEMS. The problems encountered with steam turbine governing systems may be divided into four types, listed in their probability of occurrence:

a. Speed hunt (oscillation)

b. Governor bobble (rapid oscillation)

c. Hydraulic relay performance

d. Speed wander (drift)

231-8.20.1.1 Each of these problems has its characteristics and corrections. It is important to recognize their characteristics and to pinpoint the cause.

231-8.20.2 SPEED HUNT (OSCILLATION). A speed hunt is a periodic variation in turbine speed. The speed variation will generally have a frequency of 1-1/2 to 2 cycles per second. A speed hunt can easily be distinguished from other governing problems by the following:

a. Governor bobble occurs at a much faster frequency.

b. Relay instability does not appear except under a quick disturbance and is accompanied by violent linkage chatter.

c. Speed wander will be a slower instability and generally have an erratic cycle.

231-8.20.2.1 Excessive friction is probably the most significant factor in turbine speed hunt, if the turbine is operating on design steam conditions. Excessive friction can be present both in the external governing linkage and in the hydraulic relays. If governing linkage is suspected, do the following.

1. Remove all paint and dirt at pivot points.
2. Give all grease-type lubrication points an application of grease according to DOD-G-24508, Grease, High Performance, Multipurpose (Metric).
3. Oil pivot points in the linkage that do not have grease points and are not located over hot steam parts.
4. When the turbine is shut down, remove, clean, and lubricate all pivot points.

5. Check for valves binding by disconnecting the lift rods from the rest of the control system. Lift the lift rods and valve assembly by hand to verify no binding.

231-8.20.2.2 Lost motion in the governing linkage is another frequent cause of speed hunt. When inspecting linkage for excessive friction, note looseness at pivot points. Such looseness is generally confined to primary or secondary relay restoring linkage. Correct lost motion by replacing worn parts.

231-8.20.2.3 Many control assemblies have a required bias, that is, the spring force holding links and levers in one direction taking up excessive clearances. Verify proper bias.

231-8.20.3 GOVERNOR BOBBLE (RAPID OSCILLATION). A bobble is an oscillation with greater frequency than a speed hunt. Generally, the speed is steady unless the turbine has been operating for an extended period. If the turbine has been operating for some time, bobble will cause excessive wear in the entire governing linkage, which may then introduce speed hunt along with the bobble. The main reason for eliminating bobble is to reduce the accelerated wear it causes in the linkage. A bobbling turbine governor, when observed throughout its speed range, will bobble faster at the high end of the speed range and slower at the low end. The following may cause governor bobble:

a. Improper alinement of primary pilot valve housing to governor drive shaft. Alinement should be within manufacturer's requirements, typically 0.003 inch total indicator reading (TIR).
b. Condition of governor drive gears. A burr or nick will produce a speed variation.
c. Worn or damaged governor bearings.
d. Squareness of governor gear contact.
e. Runout at pitch line of either the driver or the driven gear.
f. Nicks in the cutoff edges of primary pilot valve or bushing.
g. Worn dashpot or too little clearance in the dashpot, caused by improper installation of a new pilot valve or bushing.
h. Worn pilot valve strut pin.
i. Lead in the primary pilot valve.

231-8.20.4 HYDRAULIC RELAY PERFORMANCE. A hydraulic relay is a force amplifier that takes a small accurate force signal from a governor and amplifies it sufficiently to accurately position the control valves. The hydraulic relay consists of the pilot valve and hydraulic cylinder. The pilot valve's position is maintained by the movement of the hydraulic cylinder's piston. A hydraulic relay must be able to position itself correctly. Three things effect the positioning ability of a hydraulic relay:

a. Friction in the pilot valve and bushing, in the operating piston and cylinder, and in the linkage
b. Looseness or lost motion in the linkage, or lack of proper bias
c. Insufficient stiffness in the pilot valve and piston combination

231-8.20.4.1 Refer to the manufacturer's technical manual for the hydraulic relay positioning procedure.

231-8.20.5 SPEED WANDER (DRIFT). Speed wander is a slow, nonperiodic speed variation that may be caused by an excessive deadband of the governor, lost motion in the linkage, or the controller (if the turbine is equipped with a pneumatic speed control system). The turbine may also follow load changes, with the speed changing in accordance with the speed regulation line. Each of these possible causes shall be corrected as necessary.

231-8.20.6 TROUBLESHOOTING GOVERNORS. Governor control problems are often difficult to solve. Do not hesitate to ask for technical assistance. If troubleshooting a governor with no electrical load:

1. Verify that both the overspeed trip device and the trip throttle valve are operating properly. If not, troubleshoot these first.
2. Open the trip throttle valve only enough to allow the turbine to go on governor control at or just above rated speed but below trip speed. This will prevent rapid acceleration if the governor opens the control valves.
3. Station an operator at the trip throttle valve and manual trip. Do not leave turbine unattended.
4. Always operate the unit as if it had no overspeed trip device. The overspeed trip device is very reliable but, like all devices, could fail.
5. Do not hesitate to ask for technical assistance.

231-8.21 OPERATING CYLINDER REPAIR

231-8.21.1 Control oil systems or other operating cylinder bores are often found pitted, worn, or galled and therefore are unacceptable. Those operating cylinders with rings in their pistons may be worn by D mils or have their bore increased by D mils.

Where: D = the nominal diameter of the bore in inches, before needing oversized rings or a deviation tag outside the cylinder.

231-8.21.2 The bore may wear or be machined an additional D mils before resleeving is needed. A sleeve may be made of the same or compatible material and interference-fit into the bore to re-establish original diameter. Obtain local engineering guidance; a NAVSEA waiver is not required.

231-8.22 MATERIAL PROCESSES

231-8.22.1 Various material processes are described in this section. Also see paragraphs 231-8.5.3, brush plating and 231-8.15.6, plating chrome.

231-8.22.2 NITRIDING. Nitriding is used in Navy steam turbines to harden the surfaces of components to increase wear and galling resistance. It is used on some gear elements and steam valve sliding surfaces. Typically, steam valve lift rods outside diameters and their corresponding bushing inside diameters are nitrided to prevent galling, especially galling caused by foreign material falling into the clearance. Steam valves are nitrided on surfaces in close contact with other sliding surfaces. In nitriding processes, steel is exposed to gaseous ammonia or an alkali cyanide salt bath at 935 to 1050°F for 12 to 100 hours. The outer white layer is normally removed by grinding to size. The inner nitrided layer will have a surface hardness of 92-95 Rockwell N to a certain case depth and will retain the hardness to 1150°F. Corrosion resistance of stainless steels is reduced considerably by nitriding when the outer white layer is removed. There is a two-stage process where the white layer is not removed by grinding and the part is nitrided to final (drawing) size.

231-8.22.3 STELLITE. Stellite is used in Navy steam turbines to harden the contact areas of steam valves and seats to reduce steam cutting. Steam cutting allows internal leakage past the valve seat and is the main cause of steam valve failure. The valves and seats are undercut in the seating area, stellite is weld-built up over the drawing dimension, and the valves and seats are then machined to size. Stellite is also used to clad the trailing edges of the last stages of blading to help prevent erosion from water droplets impinging on the blade.

SECTION 9

PRESERVATION PROCEDURES FOR EXTENDED OVERHAULS AND SHIP INACTIVATION

231-9.1 GENERAL

231-9.1.1 The propulsion and ship service turbine generator (SSTG) turbine aspect of inactivating and activating ships is discussed in **NSTM Chapter 050, Readiness and Care of Inactive Ships** .

231-9.2 INACTIVATING PROPULSION AND SSTG TURBINES

231-9.2.1 PROCEDURES. The following procedures for inactivating and activating propulsion and SSTG turbines are in accordance with **NSTM Chapter 050** .

a. Preserving steam sides of turbine by dehumidification. Drain water from all drains. Dehumidifying turbine-condenser-steam system area is sufficient dehumidification of turbine.
b. Preserving turbine lubricating oil contacting surfaces with grade 2 preservative.
c. Locking shaft(s).
d. Dehumidifying the SSTG gear casing if one exists.
e. See paragraph 231-9.4 for additional guidance.

231-9.2.2 EXTENDED OVERHAUL. For an extended overhaul perform the following procedures:

a. Dehumidify turbine steam side. Drain water from all drains. Dehumidifying turbine-condenser assembly is sufficient dehumidification of turbine.
b. Dehumidify and inspect SSTG gears periodically for rusting and pitting.
c. For turbine control components left installed, keep oil side in wet lay-up with system oil.
d. Drain and clean SSTG coolers. Lift SSTG collector brushes, and dry windings.
e. See paragraph 231-9.4 for additional guidance.

231-9.3 ACTIVATING PROPULSION AND SSTG TURBINES

231-9.3.1 PROCEDURE.

1. Remove oil side preservatives on turbine reactivation except for those designed for removal by operation.
2. Remove steam side preservatives on turbine reactivation except for those designed for removal by operation.
3. See the equipment technical manual chapters on installation and operation for more procedures.

231-9.4 PRESERVATION

231-9.4.1 GENERAL. When manufactured, steam turbines, pipes and components and repair parts shall be preserved, packaged, and marked in accordance with the applicable contract or military specification MIL-P-17286, **Turbines and Gears, Shipboard Propulsion and Auxiliary Steaming; Packing of** . Further preservation guidance is found in MIL-P-116, Preservation, Methods of, and MIL-C-16173, **Corrosion Preventative Compound, Solvent Cutback, Cold-Application** . Turbines and pipes on nuclear-powered ships are governed by NAVSEAINST 9210.36, **Steam Plant Cleanliness** , or NAVSEA 0989-064-3000, **Cleanliness Requirements for Nuclear Propulsion Plant Maintenance by Forces Afloat** , as applicable. Conventional powered ships with steam turbines can use this section and the above documents for guidance.

231-9.4.2 STEAM SURFACES. Steam turbine surfaces that come in contact with steam are normally made of corrosion-resistant steel but still need preservation during transit and storage and draining and dehumidification during lay-up periods as follows:

a. On board ship, drain steam turbines and attached steam piping for idle periods expected to last up to 4 months, then close drains. A few days after draining, reopen drains briefly to remove accumulated moisture.

b. On board ship, for idle periods expected to last more than 4 months, drain per paragraph 231-9.4.2.a, then dehumidify. Dehumidify the turbine with the attached condenser and attached piping, blanked as necessary. Use NAVSEA S9086-HY-STM-010, NSTM Chapter 254, Condensers, Heat Exchangers, and Air Ejectors for drying procedures.

c. For steam turbines planned to be in storage 6 months or less off a ship, steam surfaces can be preserved with a compound meeting the requirements of MIL-C-16173, grade 5 (MIL-P-116, preservative P-21). This includes Tectyl 511 and Esgard PL-5 brands. As an alternative to these preservatives, dessicants and humidity indicators in MIL-P-116 may be used. With either alternative, turbines should still be stored inside a dehumidified building and should be covered by a waterproof tarp in transit and in storage.

d. For steam turbines stored off a ship for more than 6 months, refer to the long-term preservation requirements of MIL-P-17286.

e. Lift rod bushings have historically been a close clearance area of steam valves subject to seizing due to corrosion. Consult the turbine manufacturer to determine the need for periodic exercise of lift rods in lift rod bushings, and possible addition of preservatives or lubricants.

231-9.4.3 OIL SURFACES. Steam turbine and attached piping internal surfaces that come in contact with oil should be kept in wet lay-up with system oil while on board ship; those surfaces from which oil drains off can be preserved with a compound meeting the requirements of MIL-C-16173, grade 2. One brand name is Esgard PL-2. For these surfaces in storage off a ship, refer to the requirements in MIL-P-17286.

231-9.4.4 ROTATION. Steam turbines in storage do not require periodic rotation. They should not be stored connected to gears or generators if those driven components require periodic rotation. If they are found connected to driven components needing rotation, they may be rotated after verifying that shipping locks and fixtures, bearing paper and similar restraints to rotation are not installed. See paragraph 231-3.2.4.

REAR SECTION

NOTE

TECHNICAL MANUAL DEFICIENCY/EVALUATION EVALUATION REPORT (TMDER) Forms can be found at the bottom of the CD list of books. Click on the TMDER form to display the form.